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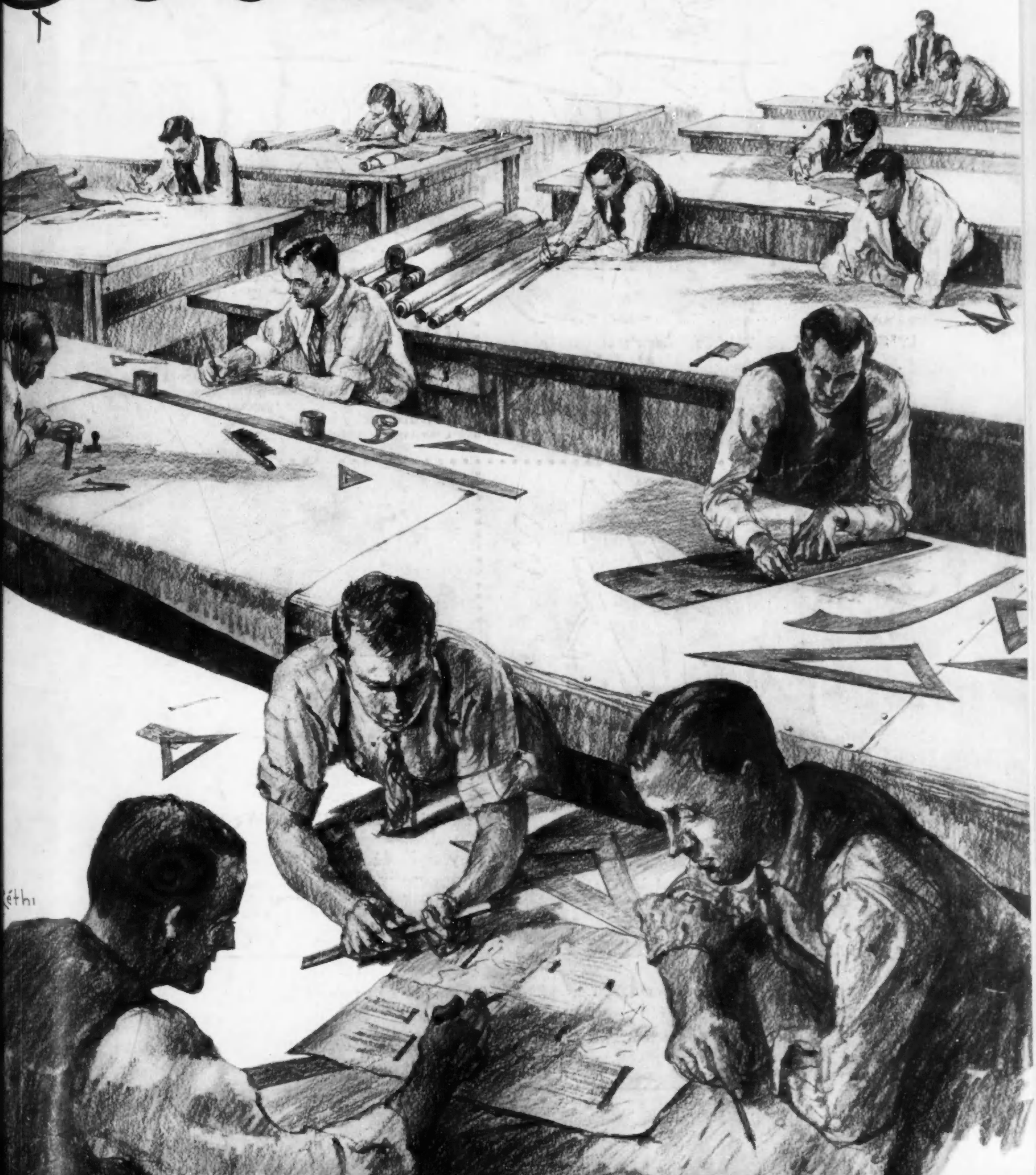
November 1949



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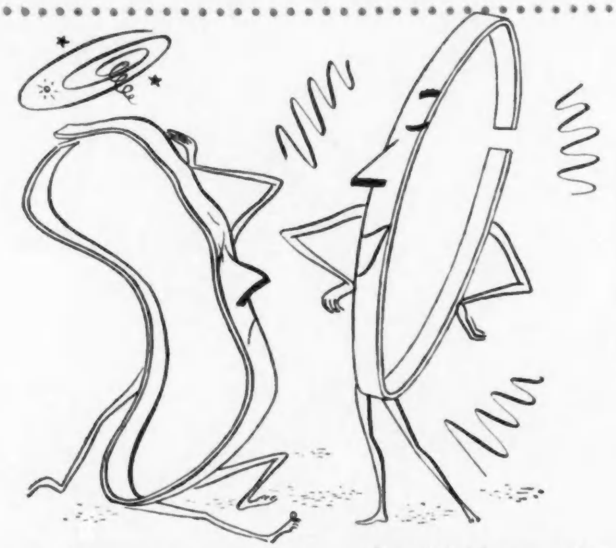
*Contributed by Evelyn Harper, 6420 Utah Avenue, Washington, D. C.**



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NOVEMBER

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Bendix Products
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OF BETTER
MOTOR VEHICLES

PRODUCIBILITY

BASED ON DISCUSSION* BY

Maj.-Gen. F. M. Hopkins, Jr., USAF

Deputy Commanding General for Operations,
Air Material Command

THIS problem of producibility is far more complicated today than it was in 1943 or 1944. Our bomber and fighter aircraft are highly complex articles resulting from the increased requirements of higher performance. The technological developments have brought about a much higher fidelity of product requiring increased dependence upon difficult fabricating techniques, manufacturing tolerances, and machining operations in critical alloys in short supply.

Those of us at Wright Field who started sponsoring producibility naturally collided head-on with those at Wright Field who were fanatical in their endeavors to get the last mile of speed out of these airplanes.

It was finally a Command decision that producibility would be considered a major item in our evaluation of new aircraft designs; that it would take its place as a major, rather than a minor, competitive item; and that it would have a point value of 20% of the total evaluation, which placed it on an equal footing percentagewise with two other major evaluation items: "tactical suitability" and "engineering."

We have divided evaluation of producibility into three groupings:

1. Design producibility.
2. Manufacturing producibility.
3. Costs.

Design Producibility

Examination of the design producibility factors allows determination of whether an aircraft is capable of rapid, efficient, expansible production. Attaining optimum design producibility will avoid the necessity for major production redesign.

It is possible to accomplish the major aims of producibility in the design stage. However, this requires an early agreement between the design and production engineer—one striving for maximum performance, the other considering processes, costs, available production equipment, and other prosaic details. This agreement can result in a high performance article capable of efficient production.

Design producibility involves simplicity of configuration, ease of fabrication, ease of assembly methods, and ease of component installation. It involves the choice of materials, the efficiency of fabricating processes, and the use of readily available tools.

For evaluation purposes, design producibility is divided into the principle factors of (1) configuration, (2) breakdown, (3) construction, and (4) expansibility. These factors are interdependent, and when properly developed, establish a logical sequence. Thus good configuration permits logical breakdown, and in turn facilitates construction, allowing rapid expansion of production.

The description of the exterior surface shape must indicate the geometric contour in terms of cylindrical, conical, circular, constant, and other sections, straight line elements, flat pattern developments, and fairing and fillet information. Production balance of these points results in simplified configuration.

As you well know, surface smoothness, gap tolerances, and joint laps are items which can complicate the production problem, hence this information must be given.

The overall production problem can be substantially influenced by interchangeability. We want to maximize interchangeability—left and right, fore and aft, and with other aircraft.

The nature and location of the principle structural members can be such that either producing these members is almost prohibitive or their location compromises or complicates producing the balance of the aircraft, or both. Consequently, we want specific information on these points.

The location of accessories and equipment must

* Discussion "Producibility" part of panel discussion on "Basic Problems of Producibility," was presented at SAE National Aeronautic Meeting, Los Angeles, Oct. 6, 1949. (Complete panel, of which this discussion is a part, is available in multilithographed form from SAE Special Publications Department. Price: 75¢ to members, $1.50 to nonmembers.)

maximize effectiveness, space utilization, and accessibility.

Evaluation of these subpoints of configuration will indicate the producibleness of the fuselage and determine whether a redesign of the airplane's configuration would be required.

Breakdown—The subpoints of breakdown are intended to establish the overall effect on producibility of the production breakdown. They should answer the questions:

1. What are the separation or assembly points?
2. What are the mating tolerances?
3. What is the sequence of assembly?
4. Are the breakdown units of a nature or size that complicate transportation or storage?
5. Are major breakdown units complete with equipment and functional components installed and can they be inspected and operationally tested prior to subsequent assembly?
6. What is the nature of attachment between these breakdown units?

In general, evaluation of this factor will determine whether efficient volume production can be accomplished without a production breakdown redesign.

Construction—Subpoints of this third factor are intended to determine the effect on producibility of the structure, the functional system, and the various joints. They should illustrate the conformance with, or the departure from normal practice. They describe the dominant features of the structure and the functional system. They define the nature of joints with information as to access for joining, clearance, tolerance, and alignment requirements; interchangeability or noninterchangeability of service and production joints; and support requirements while accomplishing the joint. Also explanation of the provisions for absorbing manufacturing tolerances through shims, oversize holes, or serrations is required.

Again, we are sifting information as to how the construction features of the design contribute to producibility. Will it require redesign to allow efficient production?

Expansibility—The fourth factor, expansibility, has a less tangible benefit in peacetime but is an absolute "must" consideration under mobilization conditions. Evaluation of this factor will determine the extent of production acceleration possible under volume production-demand conditions.

Proper planning and foresight during the design stage can minimize the requirements for unusual and time consuming fabrication techniques, for critical materials, and for complicated tooling and masters.

A bill of materials is necessary, showing rough or starting weights expressed in terms of bar, sheet, stock size, and nonstandard materials.

Manufacturing Producibility

Manufacturing producibility is divided into the factors of (1) schedules of prototype and production articles, (2) manufacturing plans, (3) plant facilities, (4) equipment facilities, and (5) manpower. The evaluation of this factor is intended to indicate whether efficient manufacture of the article is possible.

Schedules—We consider the length of time from design to prototype production go-ahead to prototype completion, and from production go-ahead to first article and to peak production. We also consider the "lead" time of government furnished property (GFP) and the "work in process" time. "Lead" time is defined as the time required between arrival of the GFP item at the contractors plant and the shop completion or final assembly of the complete aircraft. "Work in process" time is the elapsed period from placement of the material order to delivery of the aircraft.

Manufacturing Plans—We require information as to just how the aircraft is to be manufactured: the production assembly sequence, the production floor layout with flow charts, the flow times from operation to operation. This information is required for successive intervals of production from first article to peak production. Also, we need an explanation of how the unusual fabricating techniques are to be accomplished and the proposed approach to the tooling problem, indicating those aspects which would facilitate or complicate rapid production acceleration.

As you well remember from World War II, subcontracting is essential to adequate mobilization production. Consequently, we must know the items planned for off-site production.

Plant Facilities—we ask for the physical plant requirements for production of the aircraft: the flow space for fabrication and assembly, and head clearance and floor spans or "clear floor" spans—these items to be expressed in terms of on-site work. Also, we need information as to the required supporting airport facilities. These two items, plant facilities and supporting facilities, must be balanced against what the contractor has to show in the net position of the requirements.

Equipment Facilities—We again must know the net position between the production equipment required and that available. This requires the development of rather detailed information on processing with the resultant of machine loadings. Quite naturally, particular attention will be given to the requirements for complex specialized equipment.

Manpower—This final factor is intended to show the manpower required for production of the article from prototype to peak production. A breakdown is required showing allocations to engineering for production, production and tool planning, tooling manufacture, fabrication and assembly, indirect personnel, staff, and supervision. Particular information is necessary as to required know-how, available nucleus of production and supervisory personnel, rates of labor absorption, and learning curves.

Cost

The third and final major factor, cost, is very familiar to all of us. However, I believe it worthy of consideration that costs will, in the main, translate into materials, facilities, machine hours, and manpower. The information we are seeking in this factor is whether we are getting maximum return on the dollars invested.

Several points relative to producibility have been very briefly covered in this short paper. They and other points are to be thoroughly covered in the Air Force Producibility Specification.

Modifying STANDARD PROPORTIONS of Involute Gears

EXCERPTS FROM PAPER* BY

C. H. Herr

Allis-Chalmers Mfg. Co.

IN many cases, so-called standard gears selected from tables will perform very satisfactorily in transmitting power. But more and more people are becoming aware of the fact that, using the same amount of material (and sometimes even less) and the same cutting tools, the useful service life of these gears can be materially increased by modification of the gear teeth. This is important because it can be done at no increase in manufacturing costs.

Modification is the term applied to gears which do not conform to the standard gear proportions, and this modification should be done if the design can be improved. However, it is always advisable to use standard hobs and gear shaper cutters if at all possible. In other words, the modification should be done by varying the position of the cutting tool in relation to the gear blank. While it is seldom thought of in that manner, gear teeth are actually modified when they are cut thinner than standard to provide blacklash.

Other reasons for modification of gear teeth are to increase strength by eliminating undercut and providing thicker teeth, to give more continuous tooth action by eliminating undercut, and to make gears to run on nonstandard centers, in cases where these centers have already been established by other factors. In the majority of cases, however, modification is used to increase tooth strength.

Before continuing on the subject of modification, let us examine briefly the properties of the involute curve. An involute of a circle is the curve described by the end of a string as it is unwound from that circle. In the case of a gear, this circle is called the base circle. The rise of an involute cam on a line tangent to the base circle would be equal to the movement of a point on the circumference of the base circle. Thus if a point on the base circle moved through an arc of 90 deg, the cam would have a rise equal to one-fourth of the circumference of the base circle. In Fig. 1, the cam has risen a distance DE, which is equal to a distance AC on the circumference of the base circle.

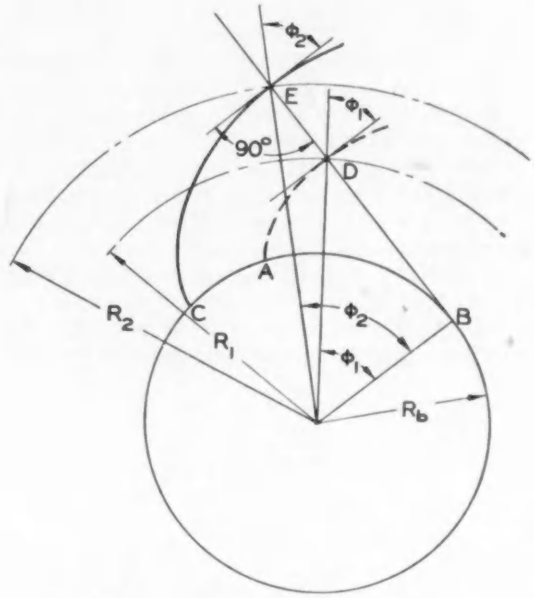


Fig. 1—Sketch showing elements determining pressure angle of an involute gear

The pressure angle at a given point on the involute curve is the angle between a line through that point and the center of the base circle, and a line tangent to the involute at that point. Thus, in Fig. 1, the pressure angle at D is ϕ_1, and the pressure angle at E is ϕ_2. Also, the line tangent to the involute at a given point is at right angles to the line through that point and tangent to the base circle.

* Paper "Modifying Standard Proportions of Involute Gears," was presented at SAE Central Illinois Section, Peoria, Feb. 21, 1949. (This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

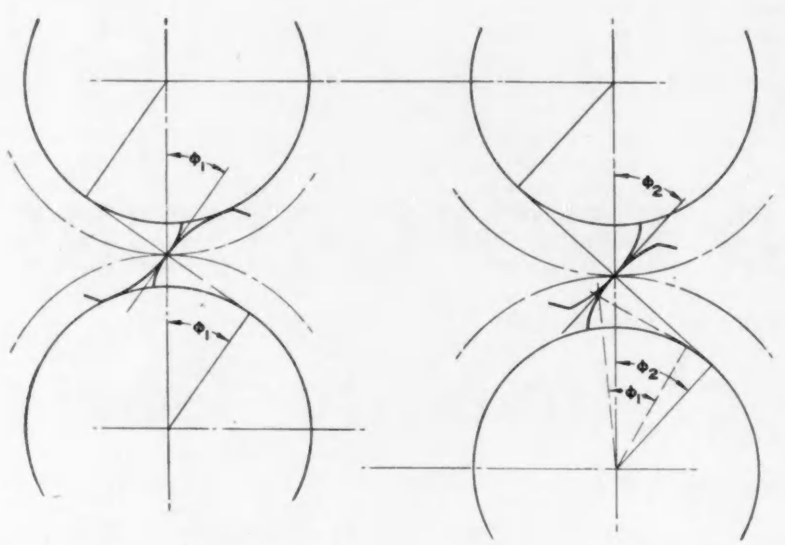


Fig. 2—This diagram shows that pressure angle increases as center distance increases, and decreases as center distance decreases

unless specifically stated at some other radius.

Knowing the pitch radius of a gear, and its pressure angle, the base radius can be found by multiplying the pitch radius by the cosine of the pressure angle, or $R_b = R_1 \cos \phi_1$. This radius determines the base circle from which the involute of that particular gear is described. Once the base circle is found, the pressure angle at any other radius can be found.

It can be seen, then, that as a point moves farther out on the involute, the pressure angle increasses.

The radius R_1, passing through the point D, is the pitch radius for the pressure angle ϕ_1, and the radius R_2 is the pitch radius for the pressure angle ϕ_2. Thus, the pressure angle is determined by the length of the radius intersecting the involute, and is limited only by the outside radius of the gear blank.

Since practically all gears manufactured in this country are specified by diametral pitch, or number of teeth per inch of pitch diameter, usually the first two things known about a gear are its number of teeth and its pitch diameter. This pitch diameter is found by dividing the number of teeth by the diametral pitch. A third factor is known, or must be selected, and that is the pressure angle. This pressure angle, as found on a gear specification, is given at the pitch diameter determined previously,

Thus, the pressure angle at R_2 is $\cos \phi_2 = \frac{R_b}{R_2}$. The pressure angle, then, is different at each different point on the involute, and it is for this reason that it must be specified at a certain radius. Actually, a gear has no pitch diameter or pressure angle until it is meshed with another gear or a rack.

It has been shown that the contact point on an involute is always on a line tangent to its base circle. Thus, the contact point between two mating involutes would travel along a line tangent to the base circles of the involutes. The angle between a line normal to this tangent line, and a line between the centers of the base circles is the operating pressure angle of these involutes, and is always equal on each involute.

Since the base circle of an involute does not change, once the involute is established, if the base

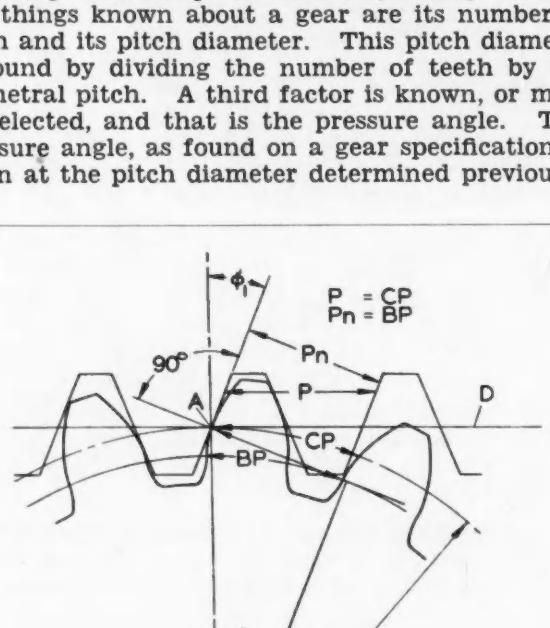


Fig. 3—A gear in mesh with its basic rack

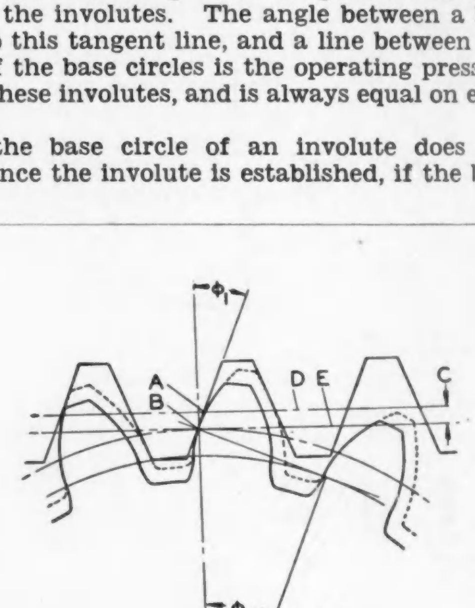


Fig. 4—Moving the gear away from its basic rack by the distance C introduces considerable backlash

circles are moved farther apart the line tangent to them assumes a different angle with respect to the line of centers; and hence the pressure angle changes.

This is shown in Fig. 2, where for one center distance there is a pressure angle ϕ_1, and for the other a pressure angle ϕ_2. This angle increases when the center distance is increased, and decreases when the center distance is decreased. This is the basis on which involute gears can be successfully operated on different center distances, for the base circle does not change when the gear centers are changed. The pitch radii, or the two radii determined by the point of intersection of the center line and the line tangent to the two base circles, will change, however, increasing with increased center distance, and decreasing with decreased centers.

The pitch radii of a pair of mating gears are in the same proportion as their respective numbers of teeth, and their sum is always equal to the center distance.

When a gear is meshed with its basic rack, though, its pitch radius and its pressure angle remain the same, even though its center is moved closer to, or farther away from, the rack. The effect is merely a change in backlash.

In Fig. 3 is shown a gear in mesh with its basic rack. It is contacting the rack at point A, and its pressure angle ϕ_1 at that point, corresponds to the pressure angle of the rack. The axial pitch of the rack, P, is equal to the circular pitch of the gear, CP, on the pitch diameter, and the normal pitch Pn of the rack is equal to the base pitch BP of the gear. The base pitch of a gear is equal to the circumference of the base circle divided by the number of teeth; and is also the spacing between two teeth measured on the line of action.

This line of action, in the case of a gear and a rack, is the line tangent to the base circle and normal to the side of the rack tooth. It is the path of the contact point between the two members when they are in motion. In the case of two mating gears, the line of action is the line tangent to the two base circles.

The base pitch of a gear is actually one of its most important dimensions. The base pitches of a pair of gears must be equal or they cannot be meshed. The accuracy with which these dimensions are held will determine the quietness and ultimate life of the gears.

In considering the action between the gear and rack, if the axial pitch P, of the rack, and the circular pitch CP of the gear are equal, the pitch radius R of the gear will roll on the pitch line D of the rack, as contact progresses from one tooth to another. The contact point A, on the rack, is located by the pitch line D on which the width of the rack tooth space is equal to the circular thickness, at the pitch radius R, of the gear tooth. For a standard gear, the tooth and space thickness are equal at R.

In Fig. 4 are shown the same gear and rack in Fig. 3. The gear has been moved away from the rack the distance C, the contact point has moved from A to B, and the rack pitch line has moved out the distance C. This introduces considerable backlash, for the pitch diameter of the gear must roll on the new pitch line E, and the tooth space of the rack at the new pitch line is larger than in Fig. 3.

However, another gear, having the same number of teeth, the same pitch diameter, and the same pressure angle as the first gear, but having a circular tooth thickness at the pitch diameter equal to the rack tooth space at the pitch line E, can be meshed with the rack without backlash. This gear is shown by the dotted lines in Fig. 4. In this gear, the outside radius and the root radius have been increased the amount C. Because the pitch diameter and pressure angle of the gear remain the same, the base circle does not change; hence the basic involute is the same, but a portion of that involute farther from the base circle is used.

If, in the preceding examples, a cutting tool, or hob, was substituted for the basic rack, it would cut the gear shown in Fig. 3, and would cut the dotted gear shown in Fig. 4. There would then be two different gears, one with larger, and consequently stronger teeth, and each one being able to operate smoothly with a common basic rack. Also, either of these gears could be meshed with any other gear having the same base pitch, the only difference being in the center distance.

It can be seen, then, that the basic operation in modifying gears is changing the relation of the basic rack, or hob, in relation to the gear center. The hob, being basically a screw thread, gashed to provide cutting edges, advances one tooth in one revolution. At the same time, through an index gear train, the gear blank is rotated one tooth space. This gives the rolling effect mentioned, with the rolling being done on the pitch circle, or generating circle, as it is called during the cutting operation, and with the gear tooth thickness at the generating circle being determined by the hob tooth space rolling on that circle.

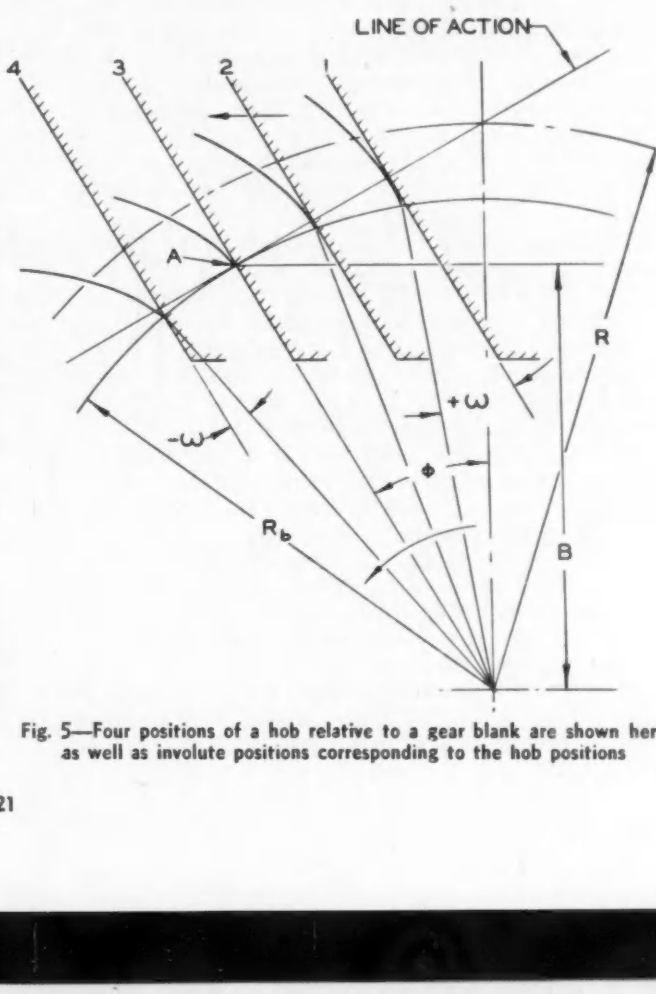


Fig. 5—Four positions of a hob relative to a gear blank are shown here as well as involute positions corresponding to the hob positions

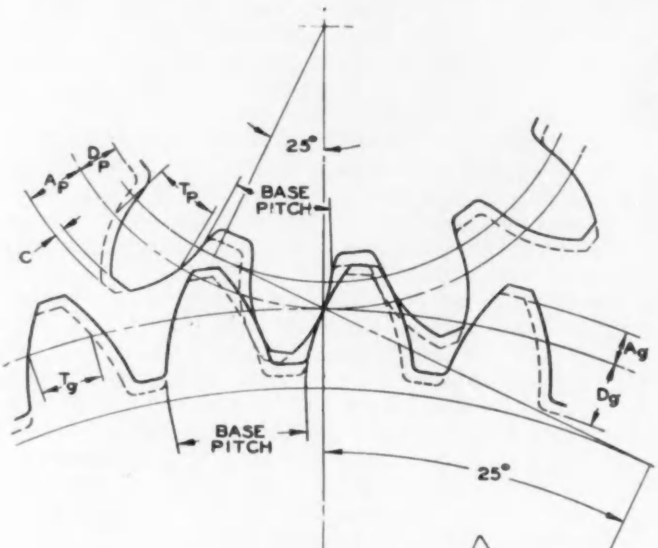


Fig. 6—Compared here are a standard pair of gears of 36/12 ratio, solid lines, with a long and short addendum pair, dotted lines

Usually, the first point to consider in modifying gears is whether undercut exists and, if so, how much movement of the hob is needed to eliminate it. In Fig. 5 are shown four positions of a hob tooth in relation to a gear blank. Also shown are the positions of the involute corresponding to the hob positions, and radial lines from the ends of the involute at the base radius R_b, to the center of the gear.

In position 1, the angle ω is formed by the side of the hob tooth and the radial line. In this position, the angle ω has a positive value. In position 2, in which the hob and the gear blank have each moved a corresponding amount, the angle ω is less, but still has a positive value. Position 3 shows the radial line at the same angle as the side of the hob tooth, and therefore, angle ω is zero.

In position 4 the hob tooth has crossed the radial line, and the angle ω becomes negative. Since the hob tooth has actually crossed the radial line, it is cutting inside that line, and will remove a portion of the involute as it continues generating. Therefore, if the hob tooth must not cross the radial line, it should not project below the point A. At that point, if the hob tooth ends and the radial line begins, the two cannot cross.

The dimension B represents the minimum distance from the tip of a sharp cornered hob tooth to the center of the gear to avoid undercut, and is found by the equation $B = R \cos^2 \phi$. Point A is called the interference point.

The most common form of gear tooth modification is the cutting of long and short addendum gears. Gears of this type are run at standard center distances, and at their original pressure angles. Fig. 6 shows the comparison between a standard pair of gears of 36/12 ratio, shown by the solid lines, and a long and short addendum pair of the same ratio, shown by the dotted lines.

The gears have a 25-deg pressure angle, and the long and short addendum pair has an addendum ratio of 0.625/0.375. This means that the addendum of the pinion is equal to 0.625 multiplied by the working depth, or sum of two standard addendums. The addendum of the gear is equal to 0.375 multiplied by the working depth. The standard pair of gears would have an addendum ratio of 0.500/0.500. Ratios greater than 0.750/0.250 are seldom used because the tip of the pinion tooth becomes too pointed.

The long and short addendum pair still has the same pitch diameters, pressure angle, and base circles as the standard pair. The difference is in the outside diameters of the gear blanks, and the tooth thickness at the pitch lines. The hob has been moved away from the pinion center a given amount, increasing the tooth thickness at the pitch line. The outside radius of the pinion is increased the same amount the hob is moved out.

The hob is then moved toward the gear center the same amount, and the outside radius of the gear is decreased likewise. The gear tooth thickness at the pitch line is then decreased the same amount the pinion tooth is increased, and the sum of the two will equal the circular pitch just the same as in the standard gears.

It can be seen, then, in Fig. 6 that the pinion tooth, which is the weaker of a pair of gears, has been increased in thickness, and consequently, strength, in the long and short addendum pair. Care must be taken to see that the gear is not undercut when the hob is moved into it. For this reason, long and short addendum gears are not recommended for ratios of less than 3 to 1.

Another type of gear modification is that used to enable gears to operate on other than standard center distances. These changed center distances may be either greater or less than standard, and will be accompanied by a corresponding change in working pressure angle, as described previously. Where nonstandard center distances are used, one gear of the pair can be standard, and the other can be modified. This is quite often done when the standard gear is already being used as part of another gear train. If only two gears are concerned, it is customary to modify both.

In Fig. 7, a pair of standard gears, Nos. 1 and 2, is shown in solid lines, with their center distance increased, and with considerable backlash. Shown by dotted lines is a modified gear, No. 3 meshing with gear No. 1 with no backlash. R_1 and R_2 are the standard pitch radii, or generating radii. WR_1 and WR_2 are the working pitch radii, and are in proportion to the number of teeth in the respective gears, and with their sum equalling the center distance.

The working circular pitch, WCP, is found by dividing the circumference of the working pitch

circle by the number of teeth, and is the same for both gears. Knowing the tooth thickness T_1 of gear No. 1, at the generating radius R_1, the tooth thickness T_2 at the working pitch radius WR_1 can be calculated. Substracting T_2 from the working circular pitch WCP, the tooth thickness T_3 of the modified gear, No. 3, at the working pitch radius WR_2 is found. T_4, the tooth thickness at the generating radius R_2 can then be calculated. From this tooth thickness, the position of the hob can be determined for cutting the modified gear. The outside radius, OR_3, of gear No. 3, is determined by the desired clearance with the root of gear No. 1.

This type of modification can be carried to the extent shown in Fig. 8. In Fig. 8b is shown a standard pair of gears of 13/26 ratio, operating on a standard 3.900 center distance. In Fig. 8a is a 12/26 ratio, and in Fig. 8c is a 14/26 ratio. These ratios also operate on the 3.900 center distance, but with different pressure angles. Also, the 26-tooth gear is the same in all three ratios.

The standard center distance for the 12/26 ratio is 3.800, and for the 14/26 ratio is 4.000. Thus Fig. 8a is an example of increasing the center distance, and Fig. 8c is an example of decreasing the center distance. The case in Fig. 8a is usually referred to as "dropping a tooth," and in Fig. 8b as "adding a tooth," and makes possible some change in ratio without changing the gear case.

While there are other possible combinations of modified gears, the results in all cases are achieved by the same means, that is, by the displacement of the cutting tool with respect to the gear blank.

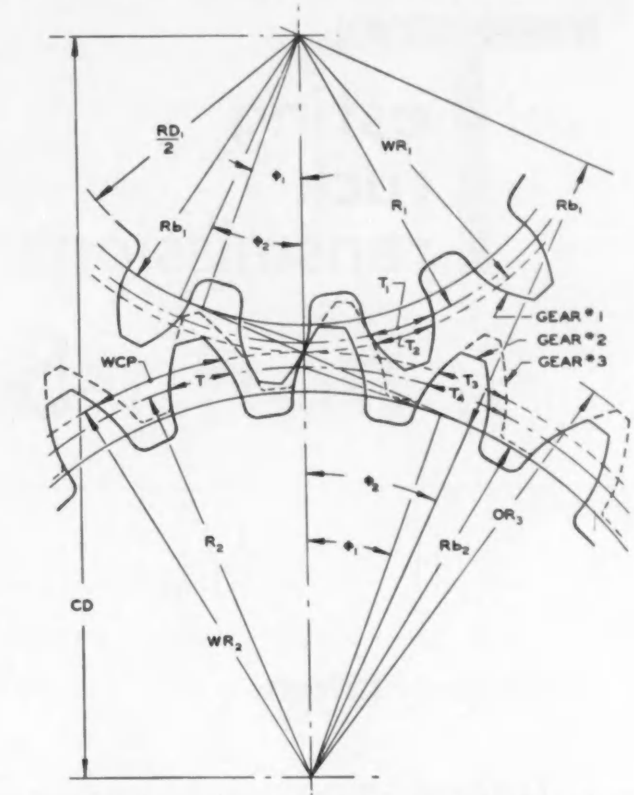


Fig. 7—Gears Nos. 1 and 2 in this case are standard gears with their center distances increased so that there is considerable backlash. No. 3, a modified gear, meshes with No. 1 with no backlash

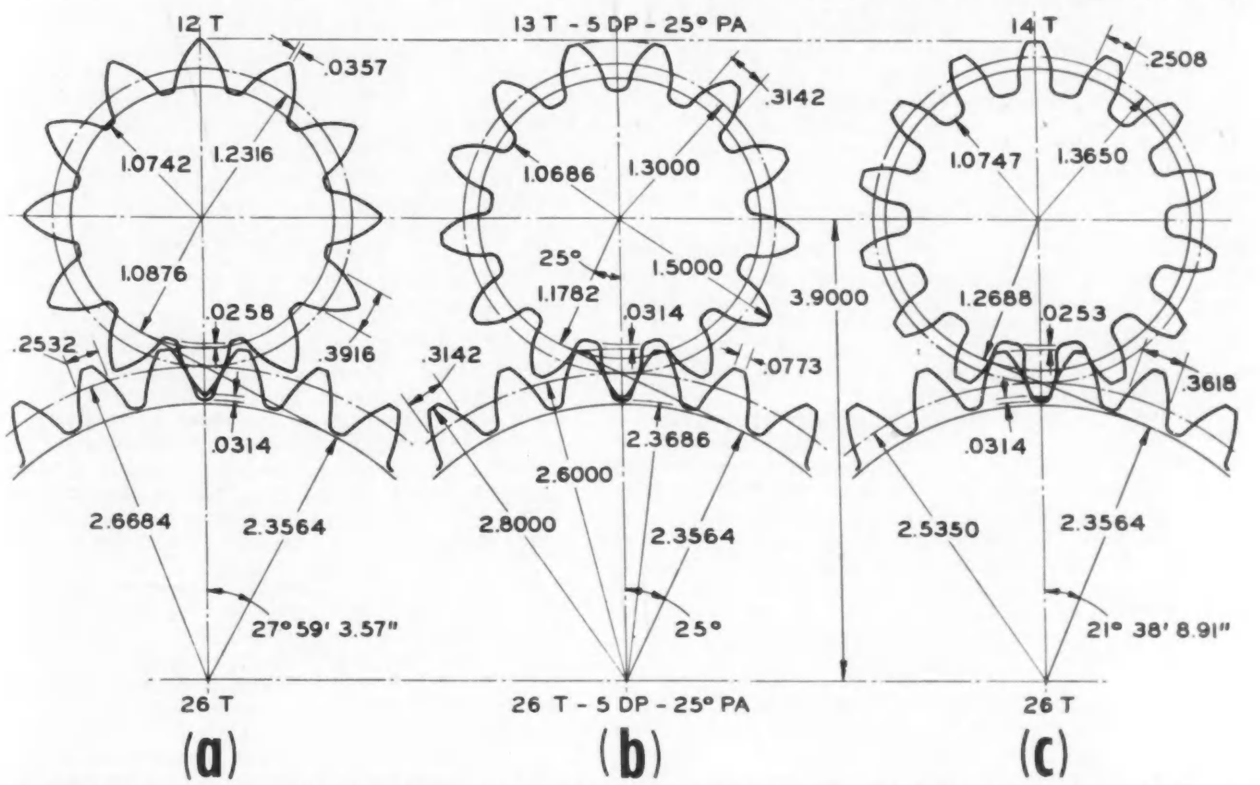
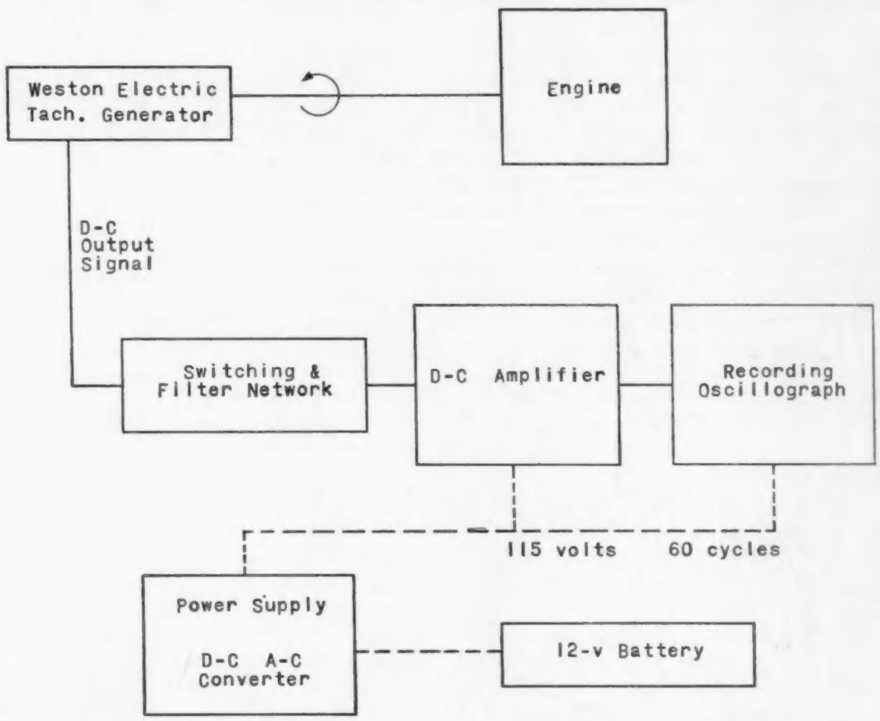


Fig. 8—The gears in (a) and (c) are modifications of the pair in (b), which is a standard pair of gears of 13/26 ratio operating on 3.900 center distance. Modification in (a) is usually called "dropping a tooth;" the ratio here is 12/26. "Adding a tooth" is the type of modification in (c), with a 14/26 ratio

Testing Truck Transmissions

Synchronized

Instrumentation for Transmission Studies



The diagram shows the instrumentation set-up used for comparing a synchronized and conventional truck transmission.

As an engine speed signal generator, a Weston Model 44 tachometer generator was used, having a direct current output proportional to speed. This tachometer was belted to an extension of the front of the crankshaft with a 2 to 1 reduction.

Since the tachometer output signal has small alternating current components, a filter was used consisting of a resistance capacitance network with a time constant of 0.020 sec. This selection of elements effectively eliminated the a-c signal without affecting the d-c signal.

The filtered d-c signal was passed through the d-c amplifier section of a Brush Strain Analyzer, model BL310S, and from there the amplified d-c signal was fed to a Brush Oscillograph, model BL-201. This oscillograph consists of a chart drive mechanism which feeds radially-ruled paper at a constant speed under the point of a pen. The pen is positioned by a pen motor which is driven by the input signal. Therefore, the oscillograph chart is a plot of variation in input signal against time or, in this case, a plot of engine speed against time.

The instrument was calibrated by running the truck engine at various speeds, as read on a Hasler Hand Tachometer, and a calibration chart plotted.

All tests were run at a paper speed of 1 cm per sec (vertical scale). One centimeter on a horizontal scale equals approximately 350 rpm, with zero speed 2 cm up from the bottom of the chart. The 115-v 60-cycle power for the amplifier and oscillograph was provided by a 150-w vibrator converter, driven by a 12-v storage battery.

Truck used for these tests was a GMC Model AF654, with a 308.2-cu in. engine, rated at 240 lb-ft and 100 hp at 2750 rpm. Transmission used was a Spicer Model 4553, five-speed overdrive, with and without synchronizers. It has a ratio of 6.10 to 1 in first and 0.77 to 1 in fifth. Axle ratio was 8.74 to 1 and tires, 11.00 × 20.

EXCERPTS FROM PAPER* BY **D. D. Robertson**
Sales Manager, Spicer Mfg. Division, DANA CORP.

Versus Conventional

OSCILLOGRAPH studies point up operating advantages of synchronized over conventional truck transmissions by showing exactly what happens during the shifting cycle.

This work consisted of plotting engine speed against time. Engine speed proved an excellent indicator of what goes on during the shift, particularly so with conventional transmissions since the shift depends on the relationship of engine speed to road speed. (See box for discussion of test set-up).

For an exact comparison, the same transmission was used for all tests, the synchronizers being removed for the conventional tests. All conventional shifts were made by double clutching, while the synchronized ones were with single clutching.

Recordings were made of engine acceleration and deceleration. These were used for plotting the theoretical shift diagrams for the conventional transmission, shown in Fig. 1. The chart at left shows an upshift from third to fourth gear, starting with the engine at 2520 rpm. The horizontal line at 1400 rpm indicates the engine speed corresponding to road speed in the fourth gear. A theoretically perfect shift would be made the instant the engine speed crosses this line.

It was established that shifts could be made with no clashing with speed differences of the mating parts of ±50 rpm and that acceptable shifts, with only minor clunking or clashing, could be made up to an approximate speed difference of ±100 rpm. This range is indicated by the shaded area. A shift made before point A will crash, though after some delay the engine may fall into the shift zone and the shift may be completed.

A shift made after point B will crash, and delay will cause worse crashing as the engine falls away from the shift zone. Note that the time interval permitted the driver for completing an acceptable shift is only 0.3 sec.

The view on the right of Fig. 1 shows a theoretical downshift from fourth to third gear, starting with the engine at 1400 rpm. The horizontal line at 2520 rpm indicates the engine speed corresponding to road speed in third gear.

With full throttle the engine follows the acceleration curve, reaching the shift zone at B in 0.4 sec. and passing through it to A in 0.1 sec. This time interval is so short that only with extreme skill could such a shift be made. Usually the engine is carried above the shift zone and advantage taken of the relatively longer 0.25-sec interval passing through the shift zone on the engine deceleration curve.

Demands Driver Care

Of course if the shift is timed so that the governed speed lies within the shift zone, there will be ample time for the shift after the engine hits the governor. But this requires careful calculation on the part of the driver for each gear change.

These theoretical shift curves do not apply to synchronized transmissions since the shift is made independently of engine speed and there is no time limitation.

Fig. 2 shows four sample oscillograph charts of upshifts indicating from left to right, the shifts from second to third, third to fourth, and fourth to fifth. Note the crash points on the third chart. These were noted manually by the oscillograph operator. The fourth chart was a trick shift made without declutching and without crashing. The driver watched the tachometer and engaged the shift at the exact speed necessary for a theoretical shift as described above.

It should be pointed out that there was a variation in acceleration and deceleration between shifts, and particularly between different drivers. Consideration should be given only to the shifting periods and not to the intervals between shifts.

* Paper "Synchronized Versus Conventional Transmissions in Truck Operation," was presented at SAE National West Coast Meeting, Portland, Oreg., Aug. 16, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

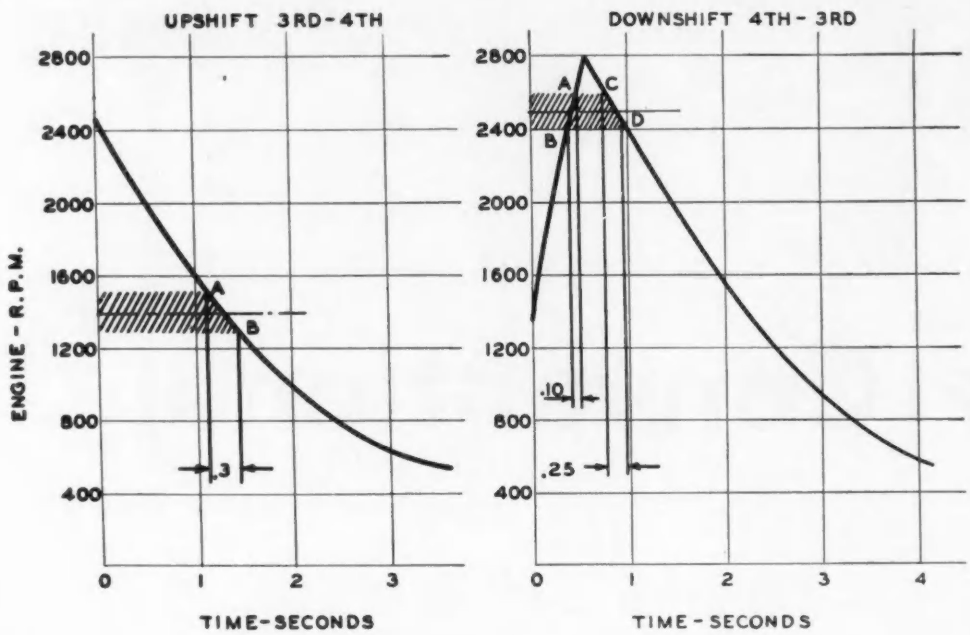


Fig. 1—Theoretical conventional shift diagrams

In Fig. 3 are shown representative upshift curves of synchronized and conventional transmissions of poor drivers. The small peak at the extreme left indicates speed-up of the engine before engaging the clutch in starting the vehicle. The points of declutching, engaging the shift, and reclutching are indicated. It should be noted that the time pattern for both types is almost identical and the shifts were made approximately at the same points. Apparently the poor driver proceeds with his shifts as fast as he can, regardless of what is going on in the transmission.

With the conventional unit, the second to third and third to fourth shifts were made too soon and the fourth to fifth too late, resulting in crashes. This is due to the wider spread between the lower speed gears and the longer time required to reach the shift point as compared with the fourth to fifth shift. Similar poorly timed shifts with the synchronizers caused no difficulty. A long interval between shift and reclutch seems typical, frequently permitting the engine to drop to very low speeds. This is shown in the sharp dips during the third to fourth shifts.

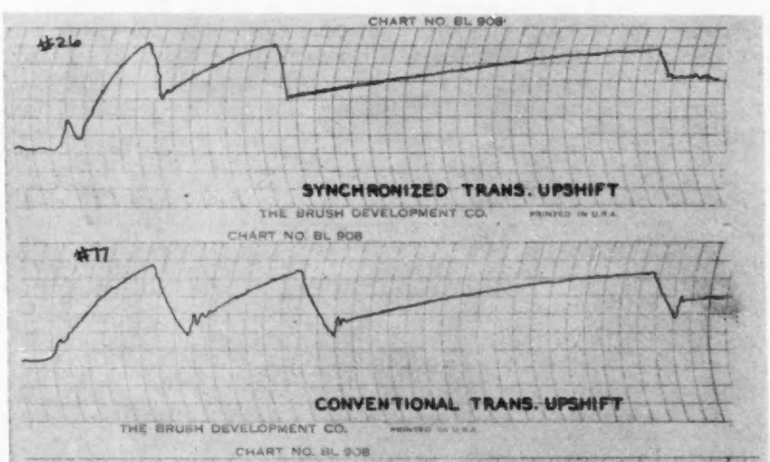


Fig. 2—Sample oscillograph charts of upshifts with both conventional and synchronized transmissions

Fig. 3—Characteristics upshift patterns of a poor driver

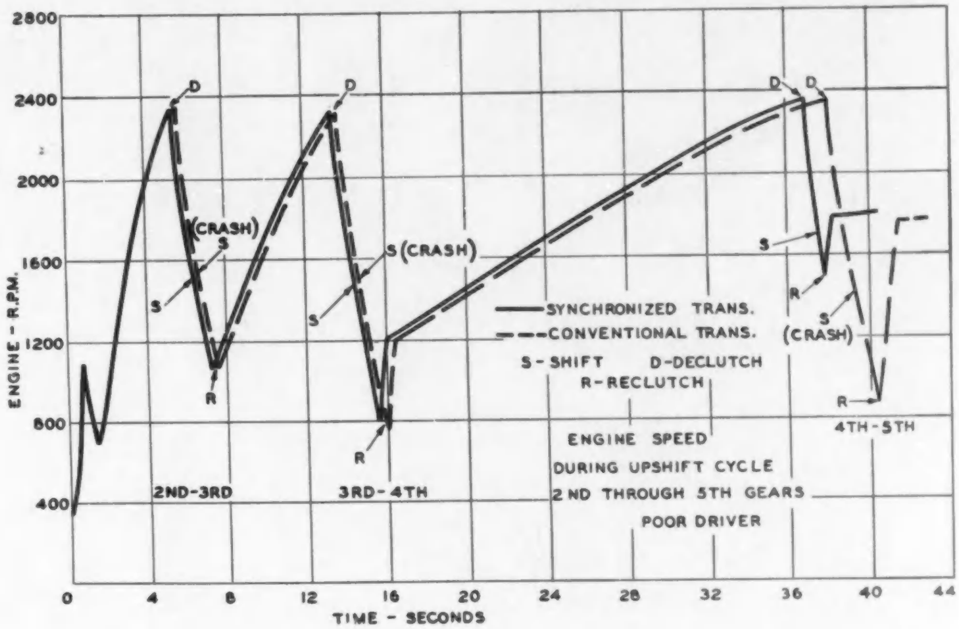


Fig. 4 is a similar comparison with good drivers. With the conventional transmission the engine is permitted to drop to the shift point before the shift is made and smooth shifts result. There is a slight further dip in the engine speed until the clutch can be reengaged and the engine brought up to road speed.

Note that there is nothing the driver can do to speed up this shift without crashing. With the synchronized unit the shift is made quickly as it is not necessary for the engine to drop down. The clutch is frequently reengaged above the shift point. The jog in the third-fourth shift curve indicates the reapplication of throttle just before reclutching pulled the engine down to road speed.

chronizers, the shift is completed during this acceleration period. With the poor driver, clutch reengagement may be made either above or below the shift zone without affecting the smoothness of the shift.

With the conventional transmission, the shift may be made during the shift zone as in the case of the fifth to fourth shift, but the curves show all too plainly the well known difficulties of the unskilled driver with downshifting. Particularly in the lower gears the shift is usually attempted well below the shift point and a crash is inevitable. Frequently a second or third crash follow. These are the most severe and serious crashes since the engine is falling rapidly away from the shift point.

Downshift Studies

Sample oscillograph charts of the more difficult down shifts are shown in Fig. 5. From left to right the shifts are fifth to fourth, fourth to third, and third to second. Upon the completion of each shift the vehicle was decelerated for the next shift instead of being accelerated until forced down by load and grade, as would be the case in normal operation. Consequently the lines between the shifts have no practical meaning.

Representative downshift curves with poor drivers are shown in Fig. 6 with the connecting lines between shifts omitted to avoid confusion.

Universal among poor drivers was a tendency to cut the throttle at the start of the downshift. This is clearly indicated on the curves. It is noticeably less pronounced with synchronizers, probably because the many motions of double clutching are not necessary. After the short dip in engine speed the engine is accelerated rapidly in an attempt to bring it up to, or above, the shift zone. With the syn-

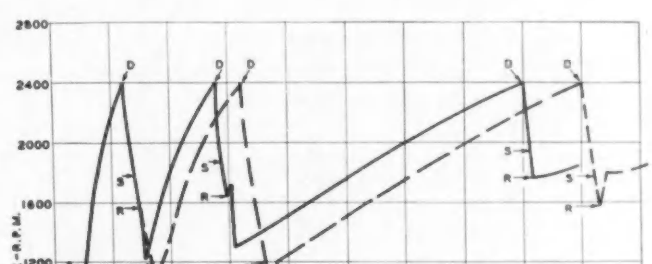


Fig. 4—Characteristic upshift patterns of a good driver

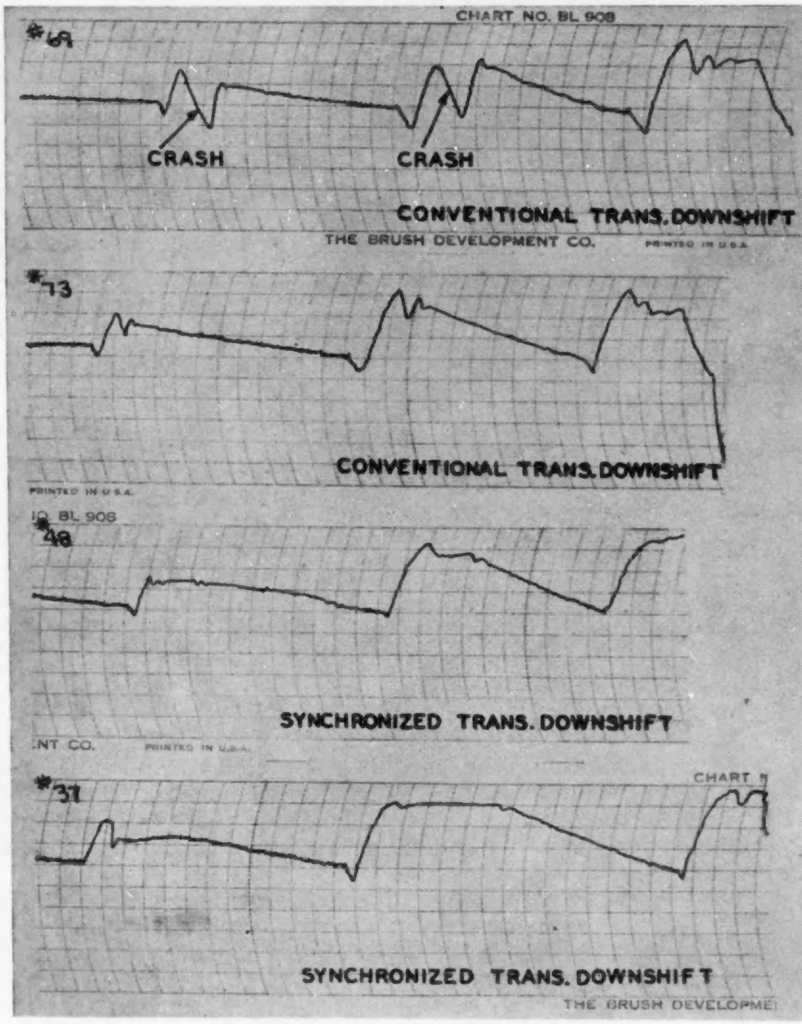


Fig. 5—Sample oscillograph charts of downshifts

If the driver is able to force such a shift, the battering and chipping of the clutching gear teeth is very serious.

In Fig. 7 are downshift curves of good drivers. With them, acceleration of the engine is begun at the start of the shift. With the conventional transmission the throttle is held wide open, usually until a speed above the shift point is reached, and the shift made as the shift point is passed coming down. A further dip takes place before the clutch can be reengaged. A great deal of skill is required for this precise timing.

With the synchronizers, the urge for instant acceleration is less since the shift is made while the engine is sped up, so that by the time the shift is completed, the engine is approximately at road speed.

A very important phase of shifting occurs when it is necessary to shift down to lower gears on steep down grades for engine braking. Missing such a shift may very well result in a runaway and a serious wreck. The factors involved in such a shift are illustrated in Fig. 8. The diagonal lines represent

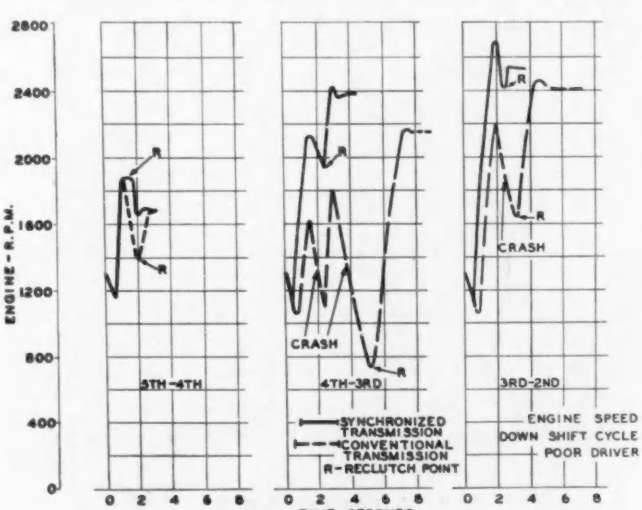


Fig. 6—Characteristic downshift patterns of a poor driver

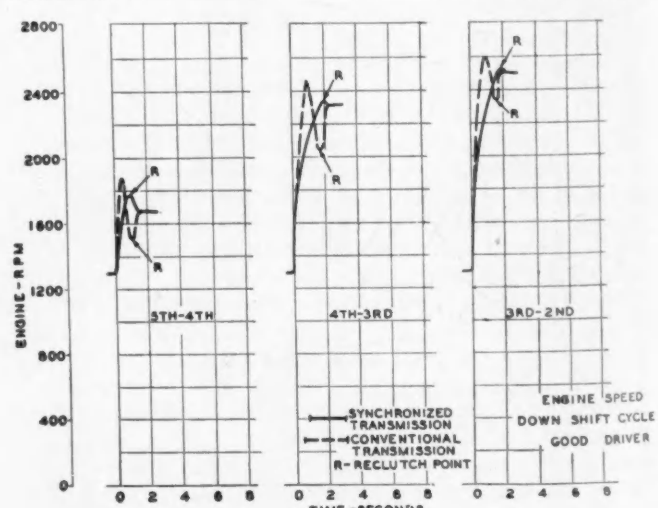
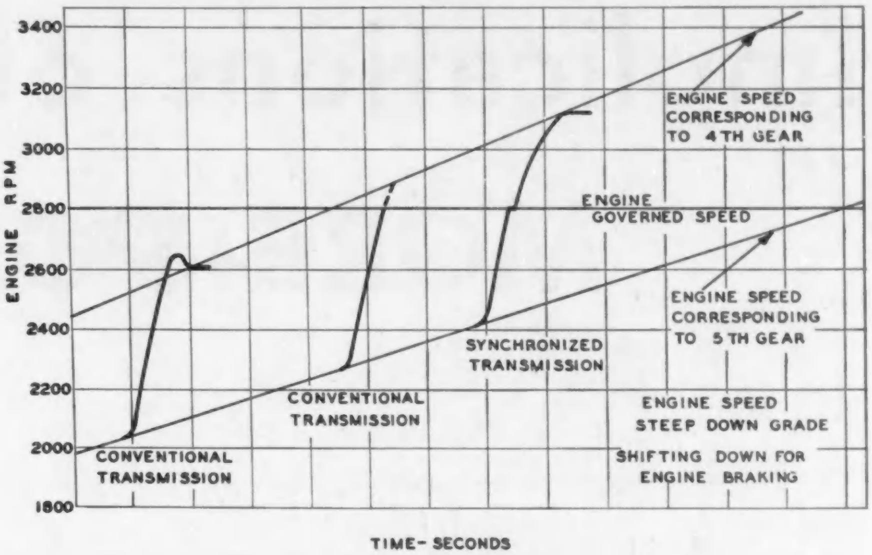


Fig. 7—Characteristic downshift patterns of a good driver

Fig. 8—Downshifts on down grades using engine braking for safety



engine speeds corresponding to road speeds in fourth gear and in fifth gear and indicate the acceleration of the vehicle on the down grade. The three curves represent free engine acceleration as taken from the engine acceleration recordings.

Assume the vehicle to be running in fifth gear. If the shift down to fourth gear is made soon enough, as indicated by the curve on the left, there is no difficulty in making the shift, since the engine speed for fourth gear is below the governor. If the shift is delayed to the point represented by the center curve, it is impossible to increase the engine speed to the shift point since this is now above governed speed. This point represents approximately the latest time it would be possible to make a crash shift and get into gear.

It is important to note that any further delay beyond this point will leave the driver in the helpless position of being unable to bring the engine speed high enough for a shift; and there is the very serious danger that the shift can not be made, leaving the vehicle running away and out of gear.

The curve on the right indicates that with the synchronized transmission it is still possible to be sure of making the shift at, or below, engine governed speed at a much later time. Reclutching will of course carry the engine up to a very high speed, but the engagement will be made and advantage taken of the braking effect.

Such a situation is obviously an emergency one. It presents an important safety feature available with synchronizers.

The large number of runs by different drivers brought out several points about shifting the two types of transmissions.

With conventional transmissions the shifting time is a function of the acceleration or deceleration of the engine and vehicle and cannot be controlled by the driver. With synchronized transmissions, as the shift is independent of engine and vehicle speed, the shifting time is under control of the driver.

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Implications of MORE-POWERFUL

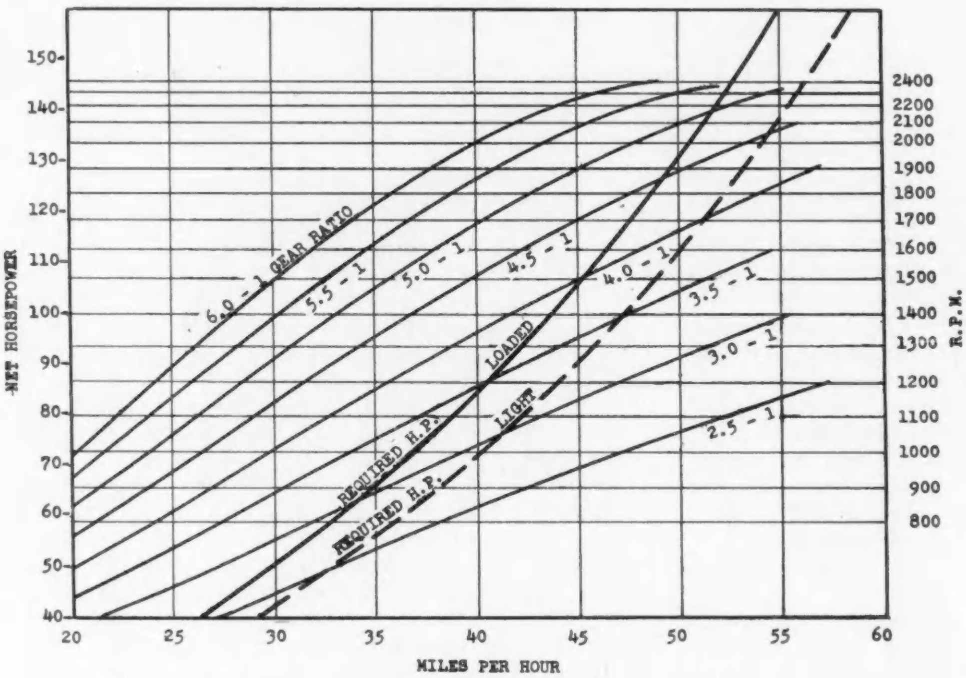

Fig. 1—Effects of gear reduction upon practicable maximum speed, loaded and light, showing that lower numerical ratios beyond that at which maximum speed is actually attainable, result in lower practicable maximum speeds. Based on a straight 4 x 2 truck of 27,000 lb maximum GVW, having a tare weight of 17,000 lb, with 11.00 x 20 tires, and an engine of 146 net hp at 2400 rpm

HOW far we should go in increasing the power of trucks to satisfy demands for higher speeds with heavier loads depends mainly on economic considerations. . . . And we may have to consider legislated performance minimums, too, now that lawmakers are concerning themselves with minimum speeds.

Back of this insistence upon more payload and speed is an economic urge. A truck earns only as it produces ton-miles; but an important part of its costs go on with time, regardless of the rate at which it produces. The greater the number of ton-miles reeled off in a day by each vehicle, the lower the unit cost. Ton-mileage can be increased in only two ways—more tons per load or more loads per day; more payload or more speed or both.

There are limitations on both loads and speeds. Loads are limited by legislative limitations on sizes and weights and also by availability of revenue traffic. Speeds are limited likewise by legal limits and by considerations of safety. Maximum daily mileage depends upon average speed. The highest average speed can be attained only by realizing the maximum practicable acceleration and maintaining the utmost speed on adverse grades.

But whether ton-mileage per vehicle-day is increased by hauling bigger loads or by increased average speeds, more horsepower is the indispensible prescription.

Obviously, though, if power increase is carried to extremes, then the margin of extra cost will go up decidedly. Our business as engineers is to determine just where the breaking point lies. It may be worth while for us to give some thought to legislation on power, too.

Here are some factors for us to consider in the trend toward more powerful engines:

Cost of Performance—That increased performance ability does increase cost is generally assumed. Certainly the more powerful vehicles, properly engineered for the power increase, will have higher purchase prices and this will affect such time charges as interest and insurance. But driver's wages, garaging, washing and polishing, painting,

TRUCK ENGINES

EXCERPTS FROM PAPER* BY

Merrill C. Horine

Sales Promotion Manager
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lubrication, inspection, and adjustment will probably remain unaffected. Similarly, mileage charges, such as amortization, engine and chassis repairs, and lubricating oil will be affected somewhat while other mileage charges, such as tires and maintenance of body and equipment should show no difference.

Even fuel, which most people assume will be consumed in greater quantity by a larger engine may actually show no increase, within limits. Experience with larger powerplants in recent years has convinced many operators that the popular notion that fuel consumption is necessarily proportional to engine displacement is erroneous. An engine is not a fuel meter. There is not much difference in the fuel consumption per ton-mile per hour between different engine sizes. Long observation leads to the conclusion that fuel consumption for a given gross weight is proportional to swept volume per mile. That this does not spell greater consumption proportional to displacement is due to the fact that with more powerful engines, the prevailing gear-reduction is less and so the swept volume does not change materially. As a matter of fact, experience of late has shown instances without number where vehicles otherwise similar and operating under like roads over identical routes, have shown greater fuel economy with the larger postwar engines than they did with the smaller prewar sizes. This is variously ascribed to the use of the slower gears a smaller part of the time, longer coasting runs due to higher speed at hillcrests, less operation on the governor due to earlier upshifts, and a better average power factor on the engine due to the same causes.

Of course, carrying power increase to extremes will boost costs markedly. Practically all items will rise except body equipment maintenance, for to carry the considerable extra weight entailed, and to conform with weight regulations, additional axles will be required which will affect such items as garage, washing and polishing, painting, lubrication, inspection and adjustment, and tires. Drivers' wages, too, will undoubtedly increase, and certainly fuel consumption will rise.

Gearing for Economy—Possibilities of broader ranges of gear ratios have hardly been explored, largely because of the knotty problems in gearbox design which have confined these ranges heretofore. Obviously for the utmost economy, the engine loading should be kept within the rather narrow speed range within which the specific fuel consumption is lowest. To accomplish this at moderate road speeds would require much lower numerical ratios than at present considered. Fig. 1 suggests some of the possibilities along this line.

Suppose, then, that we go back to the Rhode

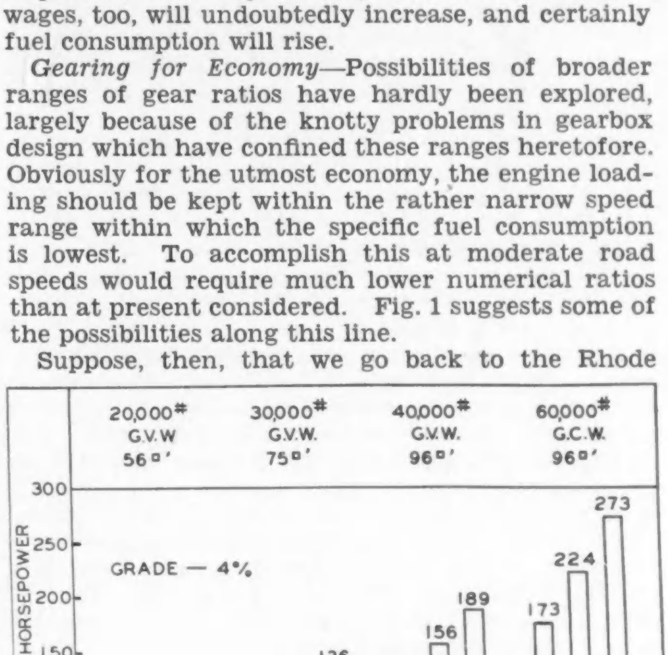

Fig. 2—Net horsepower required on 4% grade, concrete, for various gross weights at various speeds

* Paper "An Evaluation of Present Trends Toward Larger Powerplants" was presented at SAE Summer Meeting, French Lick, Ind., June 9, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

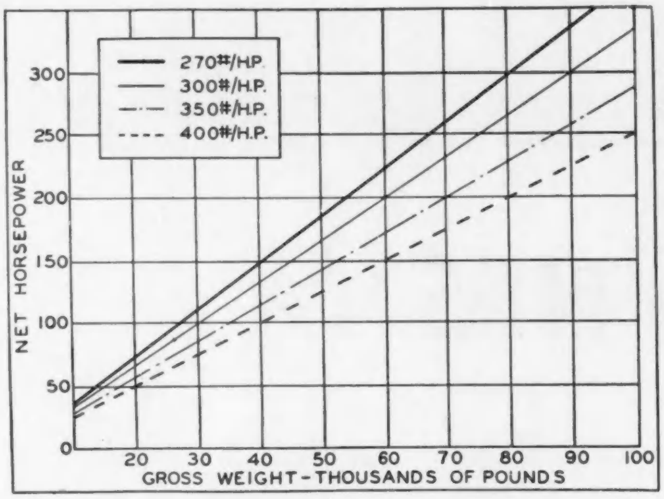

Fig. 3—Net horsepower requirements by gross weights at various weight-to-power ratios

Island idea of 4% grade ability at a certain speed. Fig. 2 shows the result. Since Rhode Island found such a requirement impossible to administer on a physical test basis, suppose we endeavor to translate these various standards into terms of pounds per horsepower. The results are found also on the chart, Fig. 2, varying from 313 to 347 lb per hp for 20 mph; from 250 to 268 lb per hp for 25 mph; and from 200 to 220 lb per hp for 30 mph.

Resorting then to the A.M.A. rating proposal, then 330, 260, and 210 lb per hp respectively would cover such requirements. Such ratios represent horsepower requirements far beyond current practice.

For a 60,000-lb combination to negotiate a 4% grade at 30 mph requires 273 net hp. As this is, furthermore, no more than the potential performance possible with such power, it presumes the exact gear ratio required for this speed with given tire size and governed engine speed. Realistically, therefore, to render such performance we would need an engine of at least 300 hp. As this is the largest truck engine commercially applicable and available for highway vehicles, this would limit gross weight to 60,000 lb.

But it would do more than that. While 300-hp engines have been adopted with apparent success by a small coterie of operators confronted with problems in many ways unique in the trucking industry, there is a much larger number of outfits on the road today handling in excess of 60,000 lb with engines rarely exceeding 150 hp. Few of these could justify the greater investment in and operating costs of such superpowered vehicles. To put legislation in force which would have the effect of obligating them either to replace their power equipment with vehicles of approximately twice their present power or to reduce proportionately their loads would drive large numbers of them out of the business and seriously cripple motor transport.

Even the relatively mild suggestion of the Highway Research Board, namely, 400 lb per hp, would bring the required horsepower of a 60,000-lb outfit up to 150 hp, as shown in Fig. 3. On a 4% grade, such a net horsepower with 60,000 lb gross would be able to sustain a speed of a trifle below 18 mph. This is a far cry from the 2 or 3 mph frequently encountered on grades no steeper than this, and for the time being should certainly be sufficient to serve the interests of highway safety.

Perhaps a saner approach to the problem of legislating for minimum performance might be a simple minimum speed limit rather than grade ability at a given speed. This would more equitably compensate for the differing requirements of different sections of the country. Maintaining a satisfactory minimum speed limit in Colorado would automatically call for more power for a given gross weight than in North Dakota. Thus the operator confining his activities to Northern Illinois, Indiana, Michigan, and Iowa would not be required to pay for outsized powerplants for the sake of the haulers in Colorado, Utah, and Idaho.

Incidentally there is a growing feeling that faster trucks are not the whole story. The higher the speed of the truck being overtaken by the pleasure car, the greater distance and time that must be spent by the latter on the left lane of the road in completing the maneuver. For example, assume that the pleasure car averages 35 mph while overtaking a 60-ft combination and that a total of 50 ft of clearance is left behind and ahead of the vehicle. Then the car must travel 110 ft to overtake the truck when the latter is stationary. At 35 mph the car will cover 51.33 fps, so it will take it a trifle over 2 sec. However, as Fig. 4 shows, as the speed of the truck increases, it takes the car a longer time and a greater distance to overtake it.

This may also be the solution of the problem of excess speed. Power enough to maintain an economic average speed or minimum speed on prevailing grades may be used to attain excessive speeds on level roads—particularly when running light. It is well known that for any combination of power, weight, and engine speed there is a breaking point in ratios. Below this breaking point, the greater the reduction, the slower the maximum speed, regulated, of course by the governor. Above this breaking point, further decrease in reduction will not produce higher speed, but lower, since the required horsepower for the theoretical speed exceeds that developed by the engine.

If this phenomenon is deliberately exploited by going beyond the breaking point so that the actual maximum speed, loaded is about 10% below the theoretical, maximum economy may be expected in normal operations, but the speed attainable empty may prove excessive. However, if there be still faster ratios beyond this point, it will be possible to restrict top speed at will, while at the same time reducing engine revolutions at the balancing point to effect still greater economy.

From the chart, Fig. 1, it will be seen that with this particular vehicle, the maximum speed attainable loaded is 53 mph with a 5.5 to 1 ratio at the governed speed of 2400 rpm. As this is the governed speed, of course, the maximum speed light, or unloaded, will be the same. With a 5.0 to 1 ratio, light, the maximum speed will be 55½ mph at 2350 rpm. Lower numerical reductions, however, steadily reduce the maximum speed in both light and loaded condition, as the curves show, until with a 2.5 to 1 ratio, light, the maximum speed would be only 35½ mph at 750 rpm of the engine.

Some day we might even govern the truck by

means of an infinitely variable automatic transmission, leaving the throttle in a fixed position and slowing the vehicle by going to an increasingly lower numerical reduction.

Variable Governors—Naturally it may be expected that as drivers discover that higher road speed is possible in the next or second speed under the top, they will select such speeds when making up time lost at an unduly protracted stay at Mom's Diner. To keep maximum speed down while enjoying the important economic benefits of low numerical reductions, various governing schemes have been experimented with. One of these employs a dual-drive governor which responds to either a set maximum engine speed or a set maximum vehicle speed, having drives from both the engine and the transmission tailshaft through overrunning clutches.

Another scheme is to have a variable speed spring in the engine governor whose tension is automatically varied in set amounts upon engagement of the two or three top gears. There is no question that, if the problem becomes really serious, practical solutions can be found.

Weight of Driving Parts—That higher-powered vehicles must be huskier throughout to withstand the greater torque loads imposed is the exaggerated prevalent impression. In the first place, torque does not necessarily increase with higher power. If high rotative speeds are the means of gaining power, then maximum torque might remain the same. Also, on the principle of Newton's Third Law, torque input will be limited by torque reaction, which will be about the same where the same mass and basic tractive resistances are involved.

Recent practice in truck design bears out these observations. If transmissions, driveshafts, and rear axles had grown in size and weight proportionally with engine power increases, present day high-powered trucks would indeed be monstrosities. Actually, for given gross ratings, trucks have not materially increased in weight, though elaboration and increased capacity of electrical, air, fuel, and cab equipment alone might have justified appreciable weight increases.

Legislating for Power—From preoccupation exclusively with maximum speeds from the standpoint of safety, lawmakers began to concern themselves with minimum speeds. In 1935, Rhode Island issued a regulation requiring 20 mph on a 4% grade; but found the requirement impracticable to enforce. In 1936, California proposed a requirement of 20 mph on grades from 3 to even 6% but abandoned the idea. In 1937, in a report to Congress, the

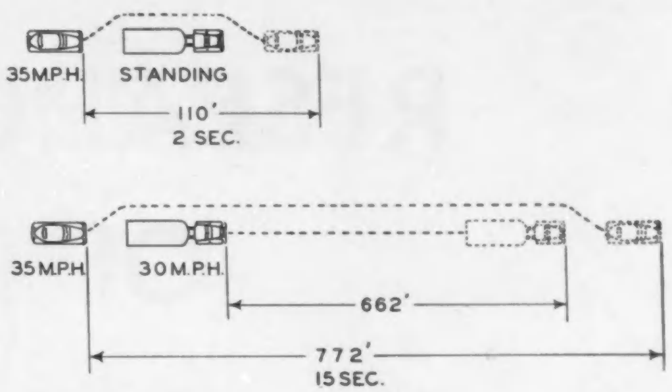

Fig. 4—Effect of truck speed upon time and distance required to overtake and pass

Federal Highway Administration pointed out the danger inherent in the large disparity in speeds between pleasure cars and heavily laden freight vehicles, particularly on grades, and demanded that truck manufacturers come up with a yardstick of performance which could be used by authorities in enforcing minimum performance limits.

This request was passed on by the Automobile Manufacturers' Association to the Society of Automotive Engineers which in 1939 recommended an ability formula of $\frac{\text{GVW}}{\text{Net hp}}$, the numerical value of the quotient or ratio to be determined by state authorities.

Accepting this, the Automobile Manufacturers' Association adopted the formula and its truck-manufacturing members agreed to so rate their products.

During the war, of course, matters remained in status quo. In 1947, however, North Carolina came out with a requirement of at least 300 cu in. of piston displacement for gross combination weights of 40,000 lb and later 350 cu in. for 50,000 lb.

At the present time, renewed discussion of the pounds-per-horsepower ratio as a qualification for licensing has brought forth conflicting demands. The American Automobile Association, spokesman for the pleasure car owner, has come out for 375 lb gross weight per hp maximum. The Pilot Study on the Highway Research Board's Project No. 5 adopted 400 lb. The Automobile Manufacturers' Association voted for 450, while the for-hire operators want at least 500. As yet, all of this is a matter of opinion; but the trend is unmistakably toward more horsepower per ton.

For complete coverage of . . .

SAE National TRACTOR MEETING

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RESEARCH REVEALS

Of Power Cranes

Fig. 1—Research on excavating machines such as these, representing one of the largest and one of the smallest shovels in standard production, is disclosing more about what happens to them in operation. The larger one, worth about $1,000,000, has a dipper capacity of 36 cu yd, as compared with the 3/8-cu yd capacity of the smaller $12,000 shovel. (To see it, focus your magnifying glass on the lower right-hand corner)

OPINION is giving way to fact-reporting instrumentation for accurately determining performance of earthmoving machines, such as those in Fig. 1, which have enough power to destroy themselves. From the stresses in selected parts of power cranes and shovels, revealed by strain gages and other testing devices, engineers are ascertaining service loads imposed on the machines.

Present excavating machine design principles stem from building machines, putting them to work, and then sitting on a bank to watch them perform. After completing a field test, it is only the observer's opinion which classifies the service as easy, normal, hard, or abusive. Establishing a reliable standard on such matters in this way requires an expensive, extensive program.

Fortunately, as much can be learned by subjecting certain parts to peak loads in a single staged test as from a large number of field tests. Method of attacking such a test program depends on the nature of the problem and the data already available for evaluating results. Bucyrus-Erie's approach to problems of loading frequency and shovel and dragline boom performance illustrates the methods for such selective tests and their results.

Loading Frequency Determination

One unknown in excavating machinery design is the general loading to which the units are subjected in normal service. Total power available plus inertia forces may develop loads considerably higher than those calculated from normal full-load power distribution; or the calculated loading may be higher than the average demand. Since any machine or structure can carry occasional overloads which will destroy it if they occur often, a design capable of carrying these overloads indefinitely will be unnecessarily heavy.

Short duration tests cannot reveal the frequency of various loading magnitudes for two reasons. First, all varying service conditions cannot be covered. Second, the operator is likely to be under tension in the presence of a test crew and may not control the machine in a normal manner. But the study of a large dragline, similar to the one in Fig. 2, over several months did point up such performance facts.

In this case the dragline owner felt that keeping records of occurence of severe loads, to avoid practices which caused them, might reduce maintenance. Equipment developed to meet this requirement also fills the need for a research instrument to measure relative frequency at which various loading magnitudes occur.

These measuring units are mechanical strain gages, shown in Fig. 3, with sufficient gage length and amplifying lever ratio to actuate precision switches, which in turn actuate electric counters to record the number of times certain loading levels occur.

One end of the reference rod is anchored to the structure being studied; the other end is attached to an amplifying lever in the switch box. This lever also is anchored to the structure at the gage length distance from the rod anchor. A spring keeps this rod under essentially constant tension. Four precision switches with repeat accuracy of a few thousandths of an inch are mounted in the box so as to be actuated by adjusting screws in the lever, which permit setting the closure of each switch to a desired stress level.

PERFORMANCE and Shovels

BASED ON PAPER* BY

Trevor Davidson and **John H. Meier**

Chief Engineer, Development and Research
Research Engineer, Bucyrus-Erie Co.

Since steel deforms elastically 0.000033 in. per inch of length under 1000 psi stress, movement at the switch contact, with a 120-in. gage length and 4 to 1 lever ratio, is 0.016 in. per 1000 psi stress. It is possible to set these gages well within this limit. In fact, for a special study on a shovel boom, these units performed satisfactorily with only 60-in. gage length.

While ambient temperature does not affect these gages, bright sunlight or other factors which may develop a temperature differential between the member and reference rod should be guarded against. A 5 F difference has an effect equivalent to 1000 psi stress.

* Paper "Research on the Performance of Power Cranes and Shovels," was presented at SAE National Tractor Meeting, Milwaukee, Sept. 13, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

For dragline installations, strain gages are mounted on each top cord of the boom and are connected to actuate a single set of four counters, because of the reversing pattern of swing load.

Fig. 4 shows readings covering two periods of about eight months each, with counters set at 93.3%, 100%, 106.7%, and 120% of calculated full load. Also shown is a chart for a three-month period for a slightly smaller machine, with counters set at 93.3%, 100%, 106.7%, and 133.3% of nominal full load. Fundamentally this is a fatigue chart and it is laid out on a log scale for the frequency of occurence.

Weakness in this data is that readings were taken at somewhat uneven intervals, with no record of the hours of actual operation. However, operation probably is on the order of 15,000 to 20,000 digging cycles per month.

The 93.3% loading occurs once in 1.5 to 8 cycles; 100% nominal loading, only once in 50 to 400 cycles;

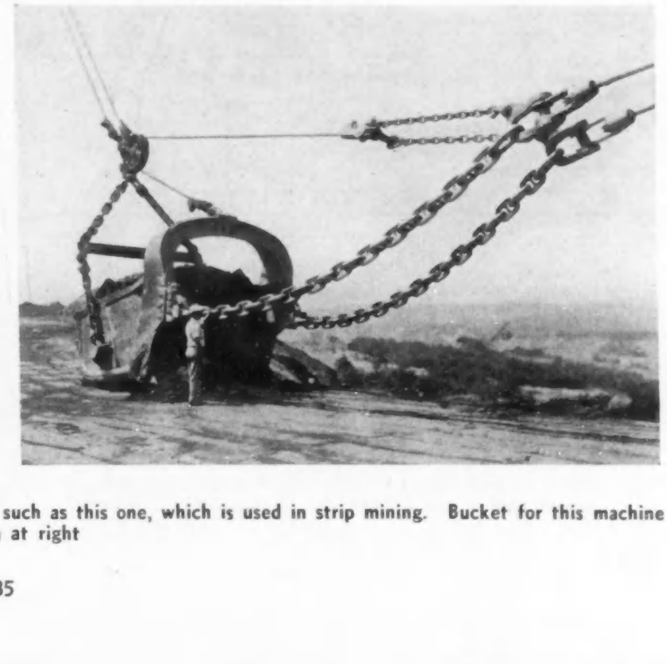


Fig. 2—Loading frequency tests have been performed on a large dragline such as this one, which is used in strip mining. Bucket for this machine is shown at right

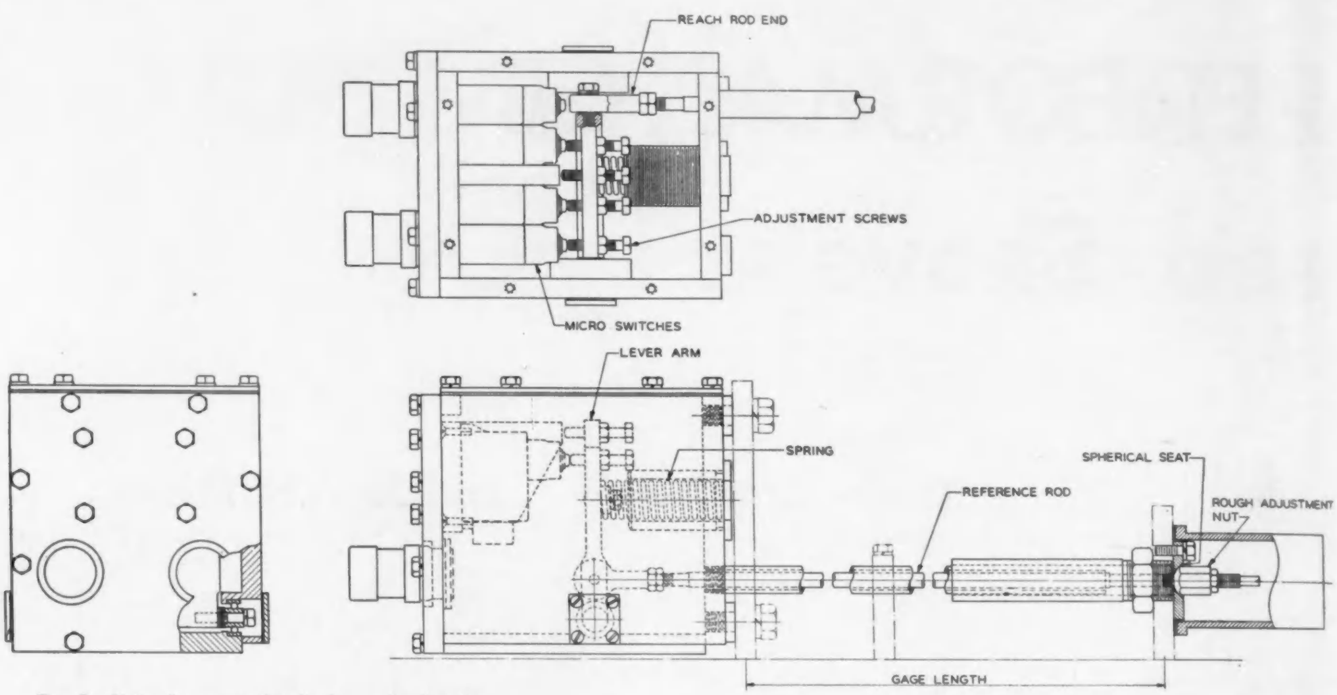


Fig. 3—Units for measuring loads on dragline booms

107% loading, once in 1500 to 200,000 cycles. The dotted lines in Fig. 4 indicate that the overload did not occur at all in some months and probably did not average more than once in 30,000 cycles. On future installations an hour meter will be added to the counter panel to show actual operating hours.

Considering that a large bucket, such as the one in Fig. 2, is digging virgin earth, the assumption that it is useless to hope for a consistent loading pattern in these machines is pardonable. This test indicates that at least with uniform digging and operating conditions, loading actually is quite consistent.

More can also be learned about shovel performance from test studies. Operational problems with a shovel stem from the fact that as the shovel digs through the bank, the dipper follows a path controlled in various degrees by the operator, the material in the bank, power and speed of the hoist and crowd motions, and the pitch of the dipper.

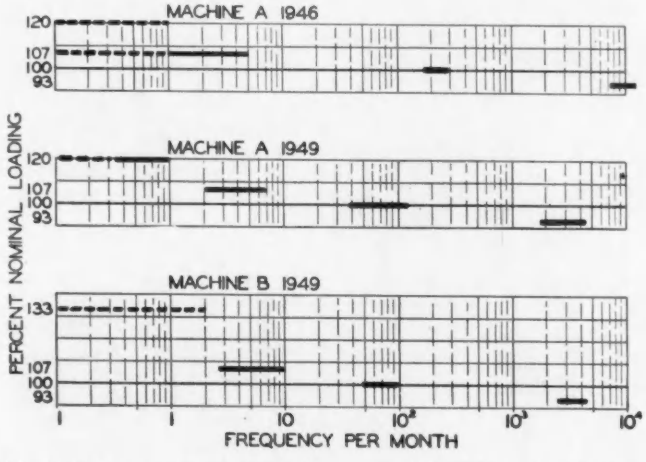


Fig. 4—These records show the frequency with which certain loadings were encountered in large dragline booms during operation

Take the case of a fairly heavy rock cut. Here the operator obviously must control the dipper to follow cleavage planes and fractures in the rock. Which cracks and fissures the dipper can enter depend on the pitch of the dipper, and the resultant digging depends on the effectiveness of power application. In digging more uniform material, such as clay, the machine characteristics may be equally important because of the operator's tendency to let the machine follow its own natural path with a minimum of control.

Dipper Action Pictured

What actually happens to the dipper in a digging cycle can be studied by the electric pantograph, shown in Fig. 5, which draws the actual path of the dipper as it moves through the bank. Working on the principle of polar coordinates, the pantograph saddle block rotates in conformity with the rotation of the shovel saddle block, and the pantograph dipper handle moves in relation to its saddle block in conformity with the motion of the shovel dipper handle.

A stylus on the end of the pantograph dipper handle traces the path of the dipper teeth by the passage of an electric current through sensitized paper. This tracing current is interrupted once a second to show tooth travel against time, which is convertible to tooth speed in feet per second. The unit is planned so that it may be adjusted for any size or type of conventional shovel.

Electric selsyn units control servo-motors to operate the pantograph. Drum "A," in Fig. 6, mounted

on the boom, is rotated by the saddle block to actuate one selsyn. A second drum, "B," mounted on the saddle block, is rotated by reciproacting movement of the dipper handle to actuate the second selsyn.

Fig. 6 is a copy of an actual trace, with shovel boom and handle drawn in to show the general relationship. The interrupted line of the dig-and-hoist path shows the travel of the dipper tooth per second. The changes in speed are apparent. The short dots of the lowering path are not on the original record, and were added here simply for clarity. The long dashes show the high speeds attained when lowering the brake.

Lowering speeds of 700 to 850 fpm at the dipper teeth have been recorded, and since these are averages for 1 sec, peak speeds may be even higher. This means the drum rotates about five or six times normal hoisting speed.

The center of the shipper shaft is located on the record before the trace is made. By laying a scale model of the dipper and handle on the trace, the angle of attack between dipper and bank may be studied.

If the weight and location of the center of gravity of the dipper and handle are known, and the hoist bail pull and crowd force measured as the trace is made, the direction and magnitude of the digging force may be calculated at any desired point on the digging path. However, both the magnitude and direction of the digging force are influenced by the momentary weight of material in the dipper, which is an unknown factor.

A reasonably satisfactory approximation seems to be to calculate two digging forces, on the assumptions that the dipper is empty and full. The actual digging force will lie somewhere between these two forces.

If the bail pull is measured at some point after the dipper is full and has left the bank, the actual weight of material in the dipper may be calculated. The average of five records taken on a 1½-yd machine digging damp sand showed the dipper to be carrying a load of 5950 lb—equivalent to 120% of rated capacity at an assumed density of 3300 lb per cu yd.

Fig. 7 shows a method of plotting power data against dipper position, which affords a convenient means for study. In this particular case the power curve represents total output torque—engine plus inertia. The power record shows two periods of high torque after the dipper left the bank. Indicators for clutch and brake applications, included on the records, show the first of these to represent the combined demands of hoisting and swing acceleration; the second represents the combined demands of further hoisting and swing deceleration or plugging.

The curve shown in Fig. 7 is not of too much practical importance, since it is one of the first records taken and was made in loose sand, more to try out the instrument than to study machine performance. But at present, it is one of the best records available.

Dragline Boom Tests

Loading counters used on a dragline boom show the stresses developed, but not the reasons for them. Several tests, using electric strain gages and recording equipment, have been run to establish the effects of operating conditions such as vertical whip and side bending. Results have not been too satisfactory because only a two-channel recording was available, limiting the data which could be taken simultaneously. There was insufficient background to plan a comprehensive test.

Tests of wider scope, using a six-channel recording, are being set up which may reveal some fundamental principles. But it is possible that an even broader program will be necessary for a thorough understanding.

Fig. 8 shows records, taken simultaneously, of swing torque and transverse bending in a dragline boom plotted against time. They were taken from a study of the effect of backlash in the swing gear.

The record represents severe plugging from full swing speed. It shows a high torque impact developed by the inertia of the swing clutch drums, which probably attain a considerable speed differ-

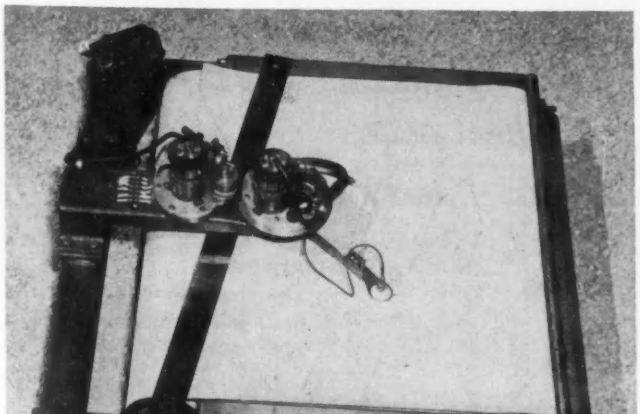


Fig. 5—This special pantograph traces the digging path of a shovel

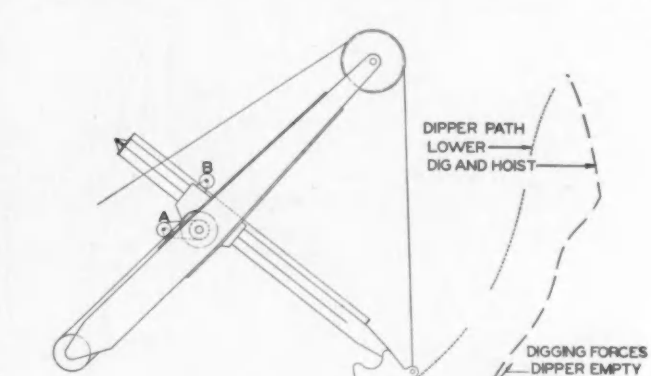


Fig. 6—Pantograph trace of a shovel's digging path, with relevant shovel parts indicated

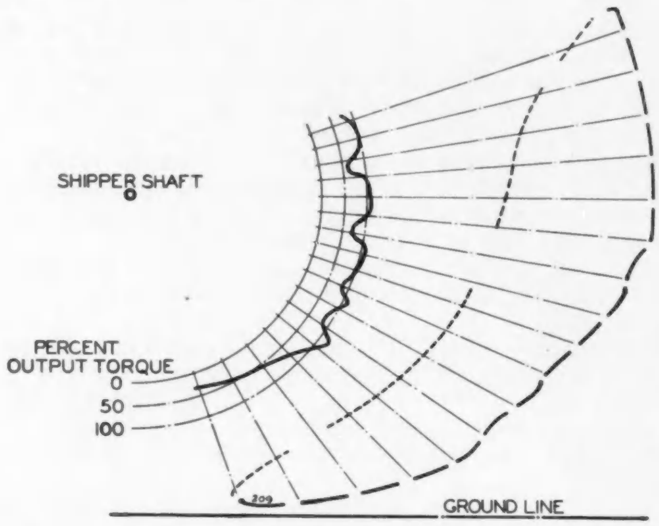


Fig. 7—Plotting power data against dipper position, as determined by the pantograph, relates these two factors, which throws more light on shovel performance

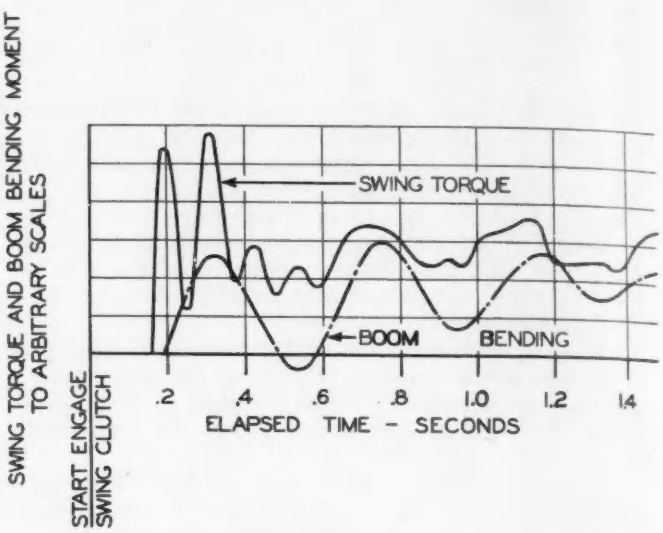


Fig. 8—Simultaneous records of swing torque and side bending in a dragline boom

ential during the time required to take up backlash. The impact is about 50% higher than the running torque. There is about 0.03 sec delay between the appearance of the swing torque and the development of side bending in the boom. The effect of sudden application of swing is to accelerate the boom so rapidly that whip develops, causing an actual reversal of stress.

It is interesting to note that the very fast vibration developed in the swing shaft is not apparent at the boom; this is because of the interposed mass of the revolving frame. But the much slower vibration in the boom is definitely reflected at the swing shaft.

Swing torque was determined from the torque in the vertical swing shaft. The installation of the torque meter, consisting of electric strain gages and collector rings, shows what can be done in special installations.

Fig. 9 shows that the swing shaft is completely covered by machinery. But by turning the shaft down about ¼ in. in diameter for a length of 3 in., there is room to mount the gages and to wax them for protection against grease. Lead wires are taken out through a hole in the shaft to collector rings at the end, the hole also being filled with wax to keep the wires from chafing. Both the weakening of the shaft and loss of bearing area under bushings are inconsequential in the short time required for tests.

When shafts reverse after a limited number of revolutions, during normal operation, and the speed is not excessive, the signal cable may be allowed to wind on and off, using hand feed if the shaft turns slowly, and some sort of retriever at higher speeds.

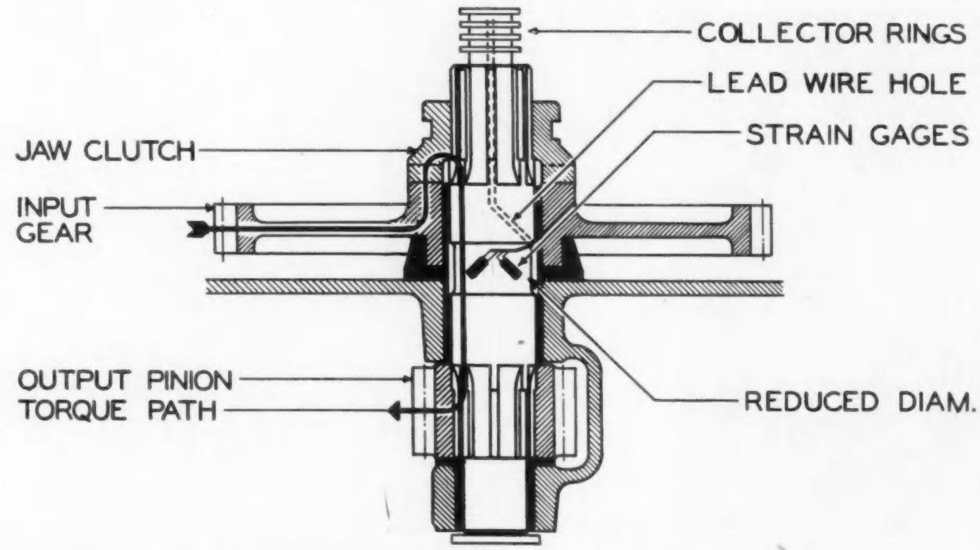


Fig. 9—Machining down the diameter of a swing shaft of a dragline made room for installation of a torque meter

Selection of Steel for Automobile Parts

What Engineers Should Know Today About Hardenability-Band Steels

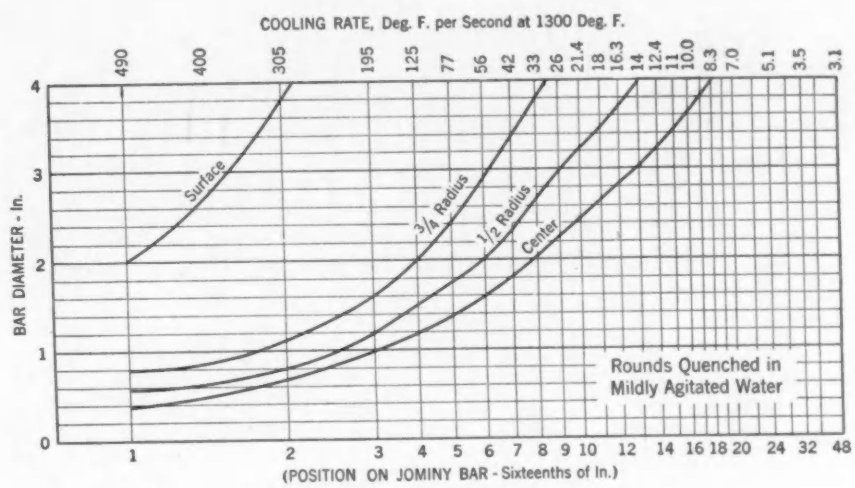
Part IV—Hardenability Selection Method

This is the fourth of a six-part report issued by the SAE Iron & Steel Technical Committee that is appearing serially in succeeding issues of the SAE Journal. The series started in the August issue. This report was prepared at the request of the SAE Iron & Steel Technical Committee's Division XVIII, Hardenability Publications. Part I was prepared by Joseph Geschelin, Chilton Co., from material provided by the Committee's Division III, Hardenability Bands. Parts II-VI were prepared for the Division by A. L. Boegehold, Research Laboratories Division, GMC.

BEFORE ordering a steel, one must know whether it will harden sufficiently in the size section for which it is to be used.

Now that we have the Jominy hardenability bar, it is no longer necessary to go through all the steps of procuring steel, machining it into sizes similar to the part to be made, quenching, sectioning and testing the hardness. Nor is it even necessary to make a Jominy hardenability bar.

All that needs to be done is to calculate hardenability from



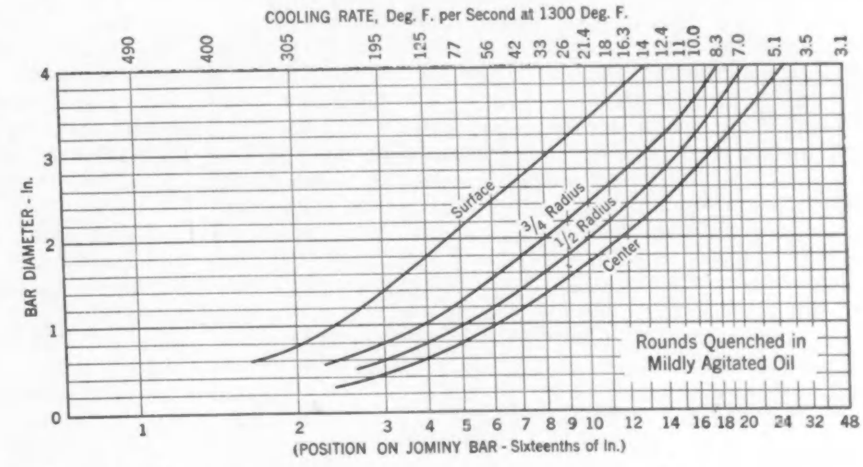


Fig. 15—Location of points in quenched rounds and in hardenability bar having the same cooling rate and thus the same hardness

Copies of the complete six-part series on Hardenability (SP-59) are available from Special Publications Department, Society of Automotive Engineers, 29 West 39th Street, New York 18, N. Y. Price: $1.25 per copy to SAE members, $2.50 to nonmembers. Quantity prices on request.

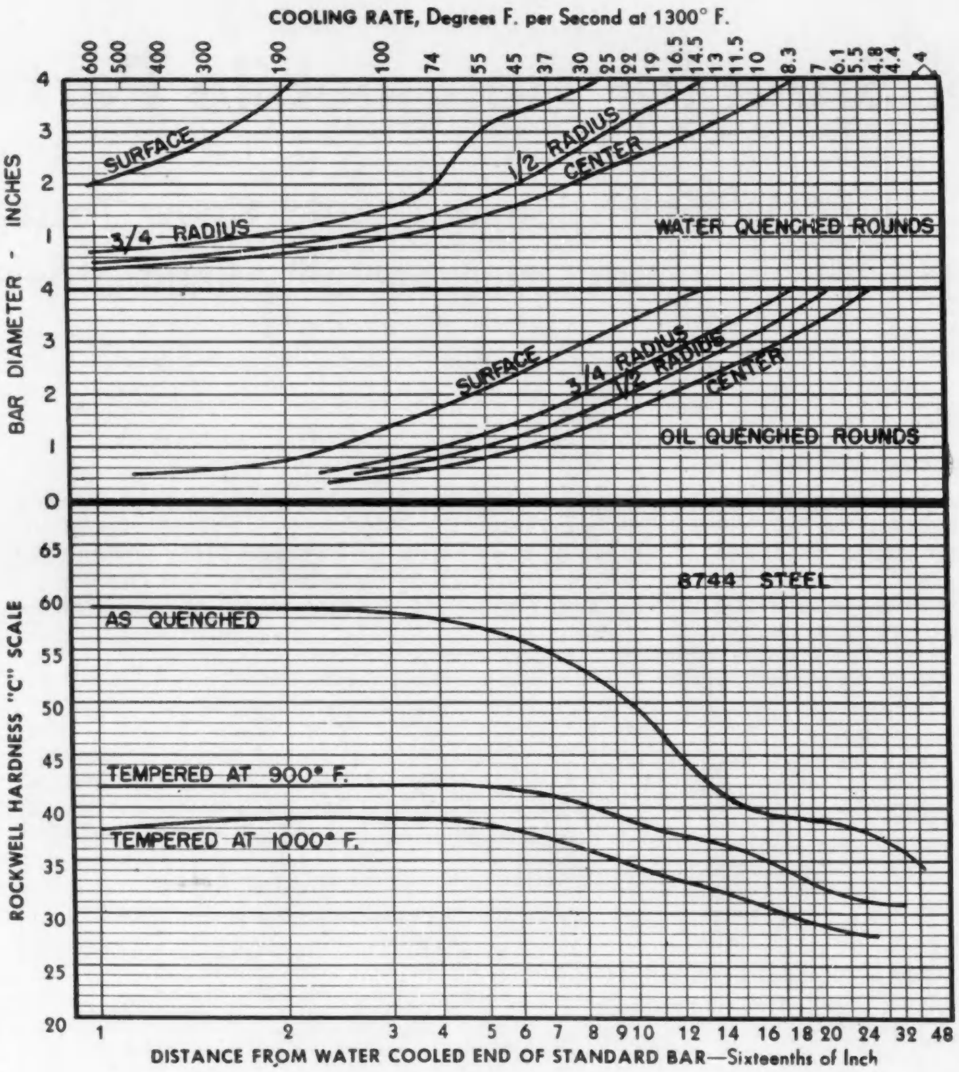


Fig. 16—Hardness in round bars determined from hardenability curve

Fig. 17—This shows how the hardenability test is used to select steel for automobile components in four steps. They are: (1) required hardness in steering knuckle is decided; (2) cooling rates of points in the steering knuckle are located on the hardenability bar; (3) required hardness values are plotted at these points on the hardenability bar to obtain the required hardenability curve; (4) hardenability curves of available steels are compared with the curve found in the third step

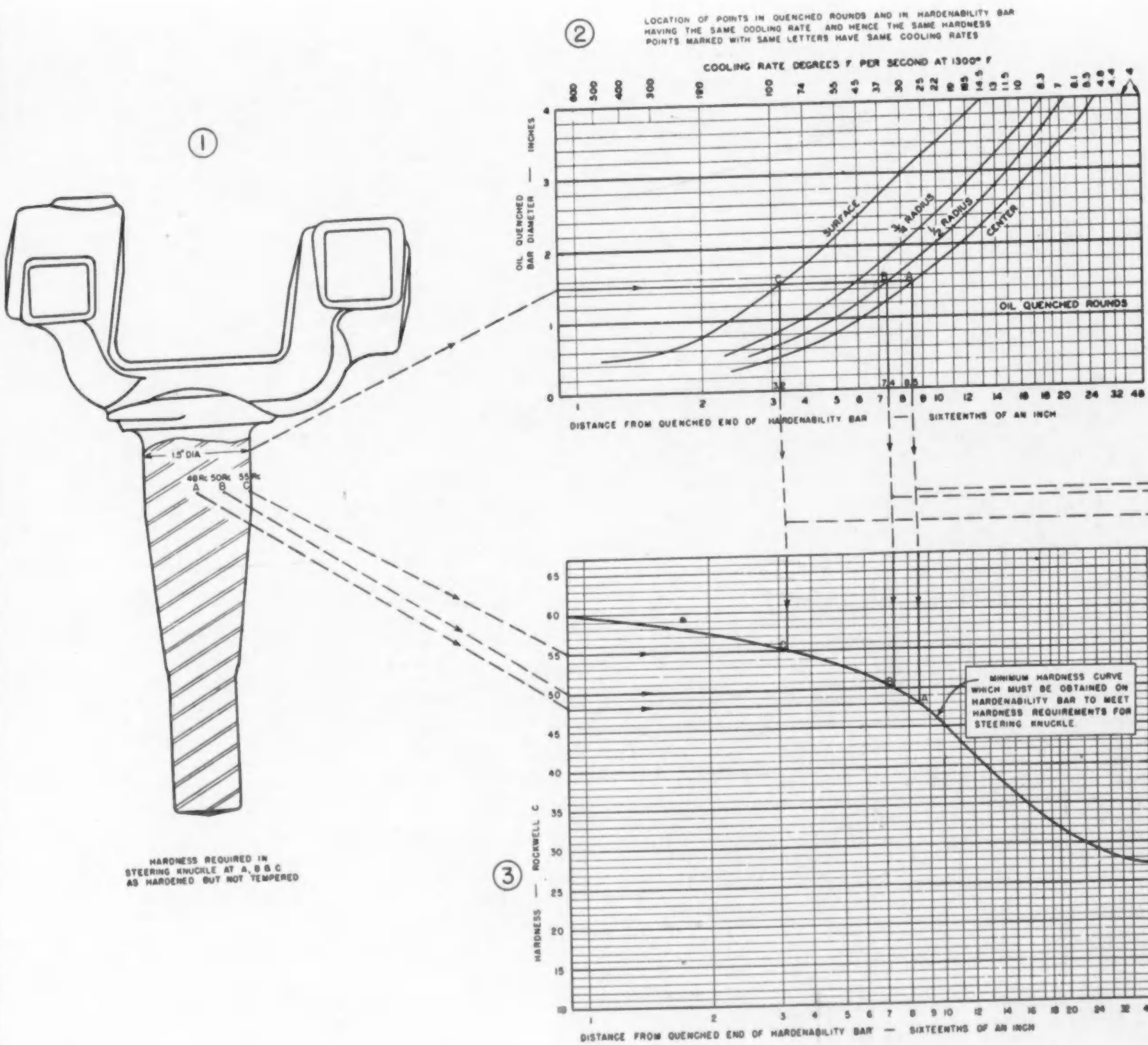
chemistry to select the hardenability curve from those provided by SAE-AISI for all commonly alloyed steels. The question is, how do we go about selecting the proper steel?

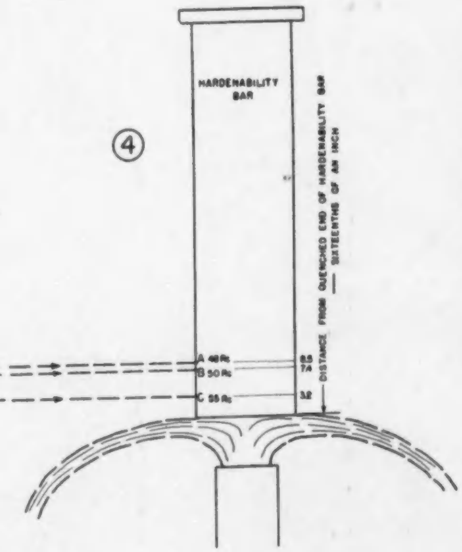
We first have to know what hardness we must have in the piece in the hardened condition, that is, before tempering. We assume that locations on the hardenability bar that have the same cooling rates as points in the cross-section of the piece we wish to make, will have the same hardness when made of the same steel. If we locate those points on the hardenability bar and ascribe to them the hardness we wish to have at the like points in the quenched article, we will have defined the hardenability curve the steel must have to meet our requirements.

To follow this through, we must have a chart which shows the location of points in quenched rounds of different sizes which have the same cooling rates as points on the hardenability bar. Such a chart is shown in Fig. 15. Where the sloping curves cross the horizontal lines representing different diameter bars, is measured the distance from the quenched end of the Jominy bar to the point which has the same cooling rate as the selected point in the quenched bar.

For example, the sloping curve for center points crosses the 3-in. diameter horizontal line at a point 17/16-in. from the watercooled end of the hardenability bar. Therefore, that point on the H bar and the center of a 3-in. round, when quenched in oil, both have the same cooling rate and would have the same hardness when made from the same steel. What the hardness would be would be determined by the shape of the hardenability curve. In the case of the curve shown in Fig. 16, the hardness would be 39 Rockwell C at 17/16-in. from the quenched end.

We have a chart, Fig. 17, that illustrates the various steps in the procedure. At the left we have the outline of a steering knuckle. We are interested in determining what steel to select for this part, so that the hardness after quenching will be not less than shown at three points in the cross-section at the critical section of the spindle where the diameter is 1½ in. Following along the line representing 1½-in. diameter in the upper chart, we find that the three points in the cross-section of the steering





THE OBJECTIVE OF WORK ON HARDENABILITY WAS A SIMPLE METHOD FOR MEASURING THE HARDENING CHARACTERISTICS OF STEEL SO THAT MANY DIFFERENT STEELS AND MANY HEATS OF THE SAME STEEL COULD BE QUICKLY AND ECONOMICALLY COMPARED.

SUCH A METHOD WAS EVOLVED USING THE END QUENCHED TEST BAR SHOWN ABOVE; THIS MET THE REQUIREMENTS FOR:

1. LOW COST.
2. AVAILABILITY FROM A WIDE RANGE OF SIZES.
3. EASY PREPARATION FOR HARDNESS TESTING.
4. INCLUSION OF COOLING RATES REPRESENTING QUENCHING CONDITIONS IN A WIDE RANGE OF SECTION SIZES.

FOLLOWING INDUSTRY'S ACCEPTANCE OF THIS METHOD OF TEST, THE NEW OBJECTIVE IS TO CORRELATE THE COOLING RATES IN THE END QUENCHED TEST BAR WITH COOLING RATES IN VARIOUS SHAPES AND SIZES QUENCHED IN VARIOUS TYPES AND SEVERITIES OF QUENCHING MEDIUMS. THIS WORK IS STILL IN PROGRESS.

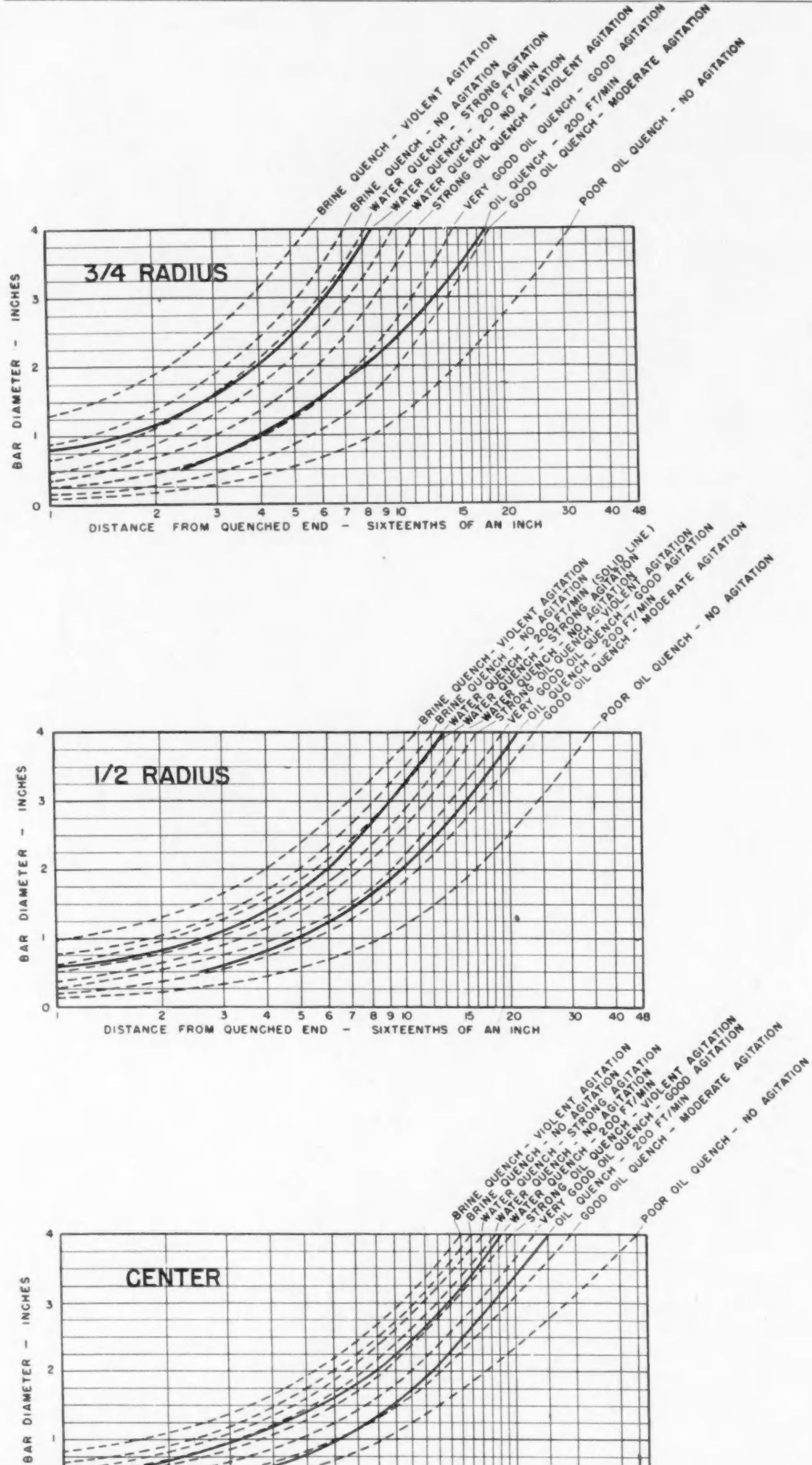
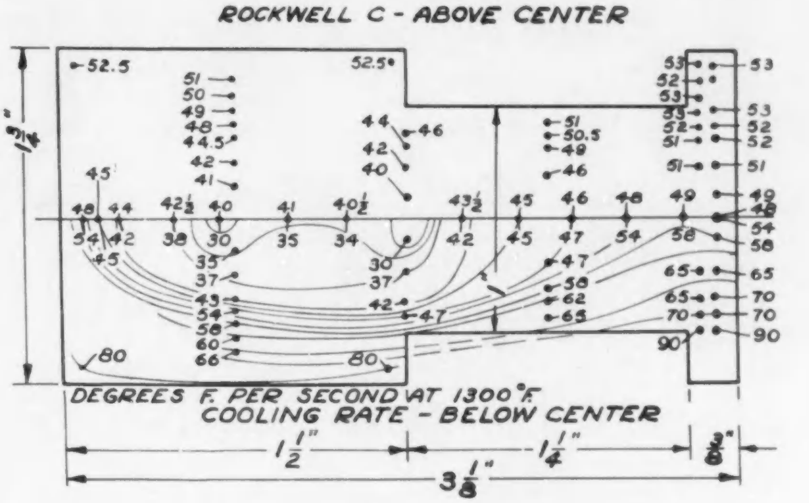
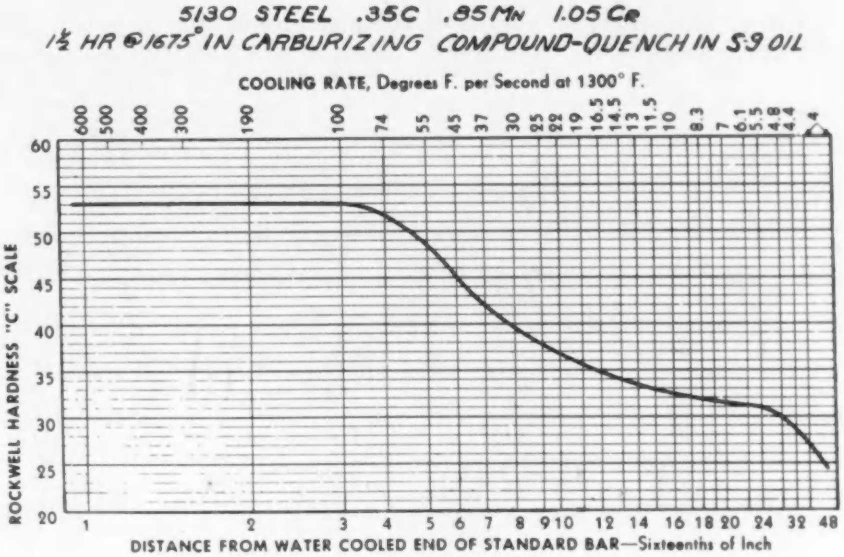


Fig. 18—Relation of quenched rounds to Jominy bar. Charts give location of identical cooling rates where hardness also will be identical (SAE Transactions, 1941, Vol. 49, pp. 266-276). Solid line—correlation from 1948 SAE Handbook; dotted line—correlation calculated by Lamont (Iron Age, Oct. 14, 1943, pp. 65-70) from works of Russell

Fig. 19—Spool-shaped part at right is quenched and hardness determined throughout longitudinal section. Cooling rates are ascribed to each position using the cooling rate-hardness relationship found in the hardenability bar. Hardenability of a test bar of SAE 5130 steel—heated to 1675 F and end-quenched—is shown in the curve





knuckle spindle have cooling rates, when the knuckle is quenched in oil, the same as occur at 3.2/16, 7.4/16, and 8.5/16 on the hardenability bar.

If we ascribe to these three locations on the Jominy bar, the hardnesses specified for the points in the knuckle that have the same cooling rates as the H bar points, we will have defined a curve which can be regarded as representing the minimum hardenability that must be in a steel to obtain the desired hardness in the knuckle. When we test a hardenability bar, like that depicted at the right, to see whether the steel has enough hardenability, we will check the hardness at 3.2/16, 7.4/16, and 8.5/16 to see whether it is up to the desired hardness. So we see that with this method, we can either predict a hardenability curve from a specified hardness cross-section curve in a quenched section, or we can predict a hardness cross-section curve from a hardenability curve.

The correlation chart shown in this display applies only to rounds quenched in oil moving past the part at 200 fpm. Any other severity of quench would require a different correlation. Cooling rates for making these correlations are not available so we are now conducting tests for determining them.

With information obtained so far we are able to project tentative curves such as shown in Fig. 18. Here are depicted the correlation curves for quenching severities, beginning with a caustic solution quench on the left, down through water quenching at various speeds, and oil quenching down to still oil on the right. With these curves we can estimate rather closely what hardness we can obtain by quenching any of these sizes up to 4-in. rounds when once we have the hardenability curve.

An alternative method for determining cooling rates in an irregularly-shaped article, especially where the quenching severity is other than those for which correlation charts are available, is to apply known hardness-cooling rate couples to locations in a quenched article where hardness has been determined. This is illustrated in Fig. 19.

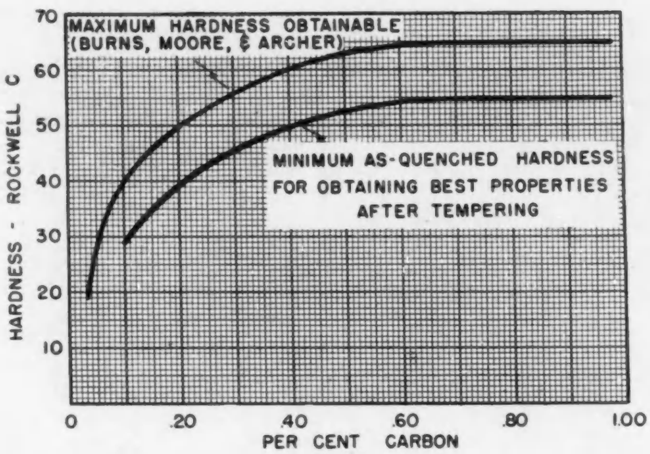


Fig. 20—As-quenched hardness for best properties after tempering

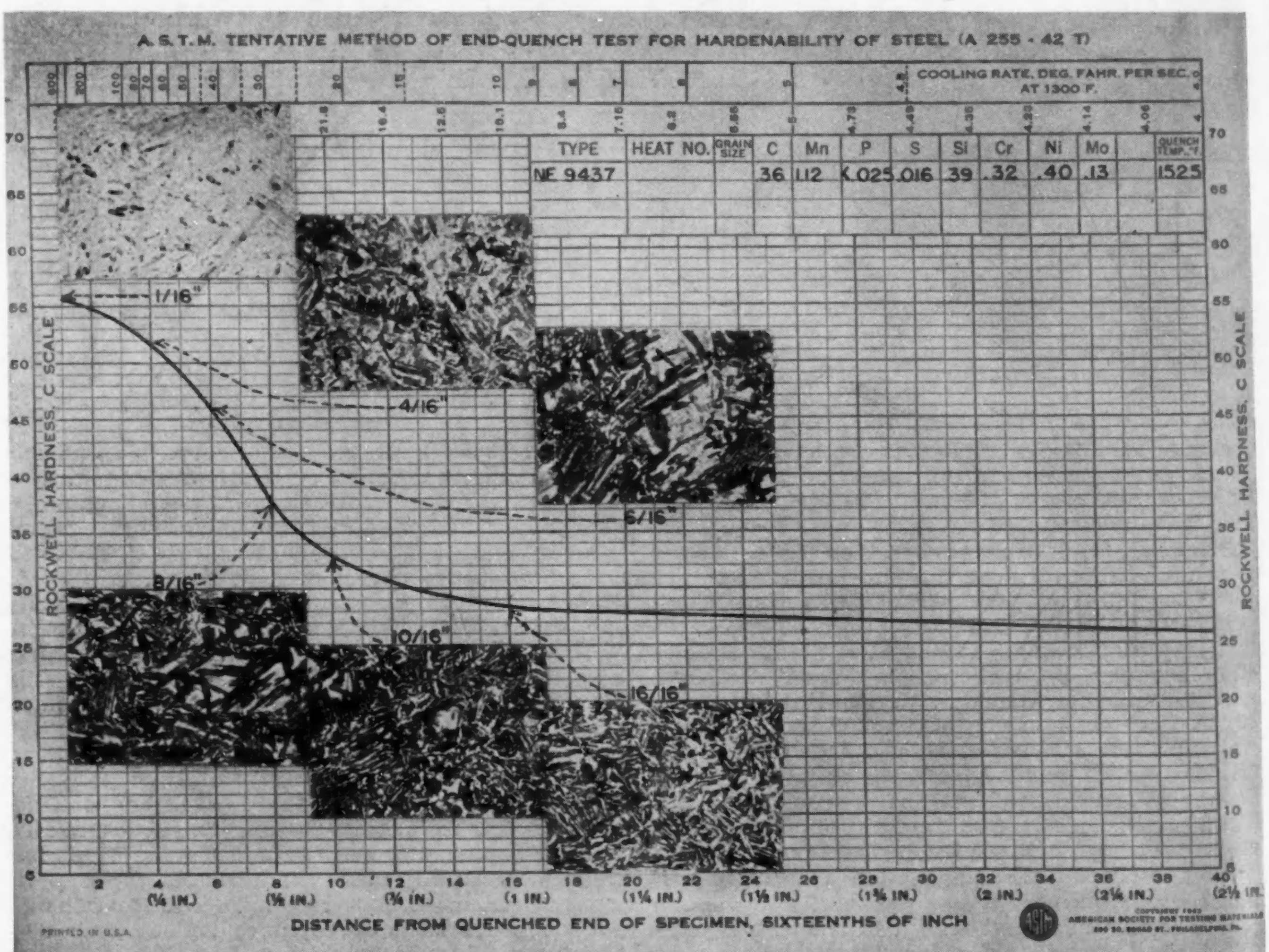
The article to be investigated is represented by the spool-shaped piece at the top. This is made of a shallow-hardening steel, and a hardenability bar is made of the same steel and a hardenability curve determined. The spool-shaped article is quenched and the hardness determined throughout the longitudinal section and cooling rates are ascribed to each position using the cooling rate-hardness relationship found in the hardenability bar. These locations also can be identified as equivalent to the number of sixteenths from the quenched end of the hardenability bar at which that hardness occurs.

Now that the quenched spool has been catalogued with respect to cooling rates, any desired hardness can be specified at various locations in the piece; and these hardnesses coupled with the cooling rates at the points specified will define a new hardenability curve which will be required of the steel to produce the hardnesses desired in the spool.

When we have established the hardenability curve required to produce the desired hardness in the part we want to make, we can compare this curve with the SAE standard hardenability curves for the various alloy steels to select the steel to specify for the part. We will bear in mind, as an added requirement, that the steel selected will be the lowest cost steel having the required hardenability and it shall have the lowest carbon compatible with obtaining the desired results.

By selecting the lowest carbon, we will obtain

Fig. 21—How microstructure varies as hardness obtained by quenching decreases



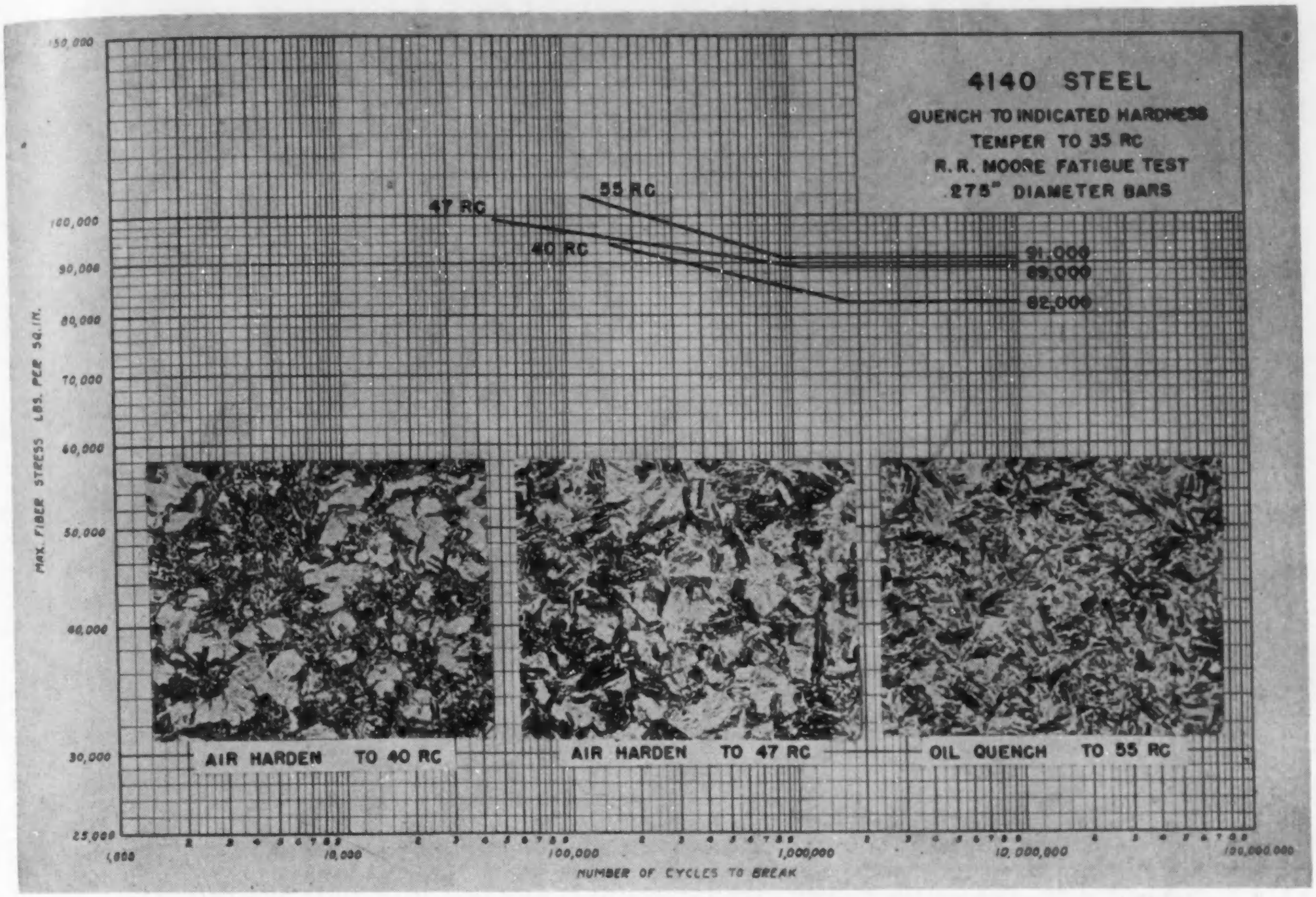


Fig. 22—How structure prior to tempering affects resistance to fatigue of steel tempered to 35 Rockwell C

lowest cost of forging and machining. Carbon is the cheapest element for obtaining hardenability when a moderate amount of alloy is present, so the desire to raise carbon to obtain hardenability and to decrease it to aid machining, work in conflict; the result must be a compromise.

In deciding what hardness is required in a hardened part, prior to tempering, we are guided by a chart in Fig. 20 which shows the maximum hardness possible for every carbon content and also the minimum hardness for obtaining good physicals after tempering. We do not adhere to the values shown in this chart in all cases. For example, in the crankshaft and connecting rods, the tempered hardness is so low that it is not important that full hardening be obtained prior to tempering. The service stresses are so low, due to the prime requisite of stiffness, that fatigue failure due to incomplete hardening is not encountered.

The minimum hardnesses shown as desired after quenching in Fig. 20 are based on the hardness obtained when the structure consists of 85 to 90% martensite. Fig. 21 shows how the microstructure varies as the hardness obtained by quenching decreases. The hardest structure is 100% martensite. The presence of bainite accompanies the next lower hardness. Bainite increases as hardness decreases; then with still lower hardness ferrite appears which increases in quantity until the hardness is that of normalized steel.

We know that the greatest resistance to fatigue in tempered steel is obtained when the quenched structure contains 90 to 100% martensite. The higher the hardness after tempering for highly stressed members, the more important it is to have full hardening before tempering.

Fig. 22 shows how structure prior to tempering affects resistance to fatigue of steel tempered to 35 Rockwell C. These tests were made on 0.275-in. diameter rotating beam fatigue test specimens. Here we see that full hardening is necessary to obtain the greatest fatigue resistance. The endurance limit of steel hardened to only 40 Rockwell C is 9000 psi less than the ones fully hardened.

Fig. 23 shows similar data for the same steel in a 1½-in. diameter test bar subject to reverse bending in one plane. This is typical of stress conditions in the spindle of a steering knuckle. Here the fully hardened specimen has the same advantage over the partially hardened, but the endurance limit is much lower in the larger size specimens. This decrease of fatigue resistance as size increases is known as the size effect and is one reason why the chart in Fig. 4 is drawn with such a wide band to indicate that the endurance limit varies as low as 0.4% of tensile strength, depending on size and other indeterminate factors.

We do not know, however, how far below the surface of a stressed member it is necessary to have maximum properties. In the case of axially stressed

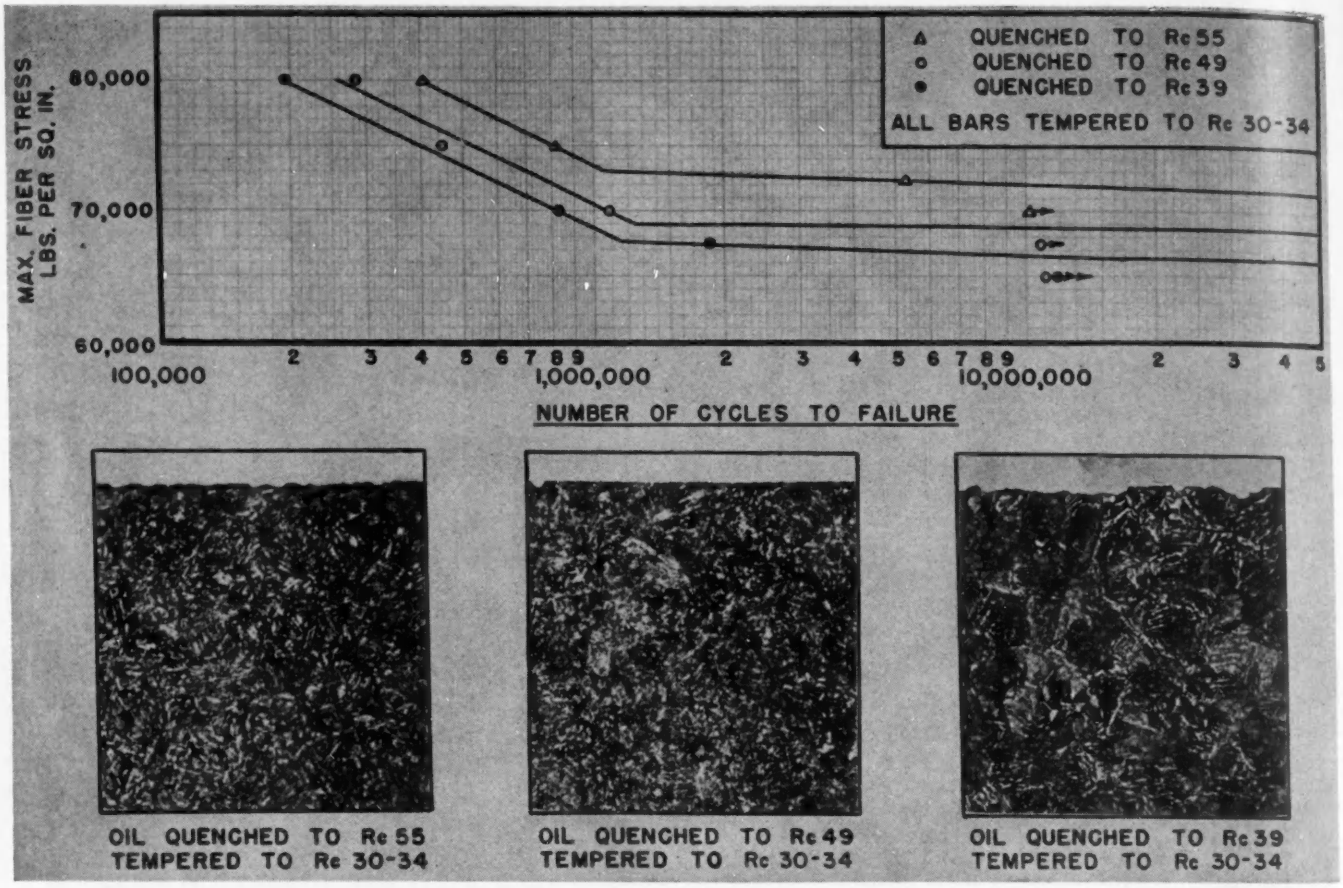


Fig. 23—These data are for a 1½-in. diameter test bar subject to reverse bending in one plane, for same steel as in Fig. 22. Note that larger size specimen has lower endurance limit

members like connecting rod bolts, it is necessary to harden fully entirely throughout the section. Where this type of hardening is necessary, the hardness that should be obtained by quenching is given in Fig. 20. The hardnesses defined by the lower curve represent a structure containing 85 to 90% martensite. In members stressed in bending, the minimum as-quenched hardness required is almost entirely a matter of determination by cut and try; that is if we wish to tolerate some degree of hardening that is less than full hardening all the way to the center of the section.

Effect of Different Practices

Depending on the design of the member and the rest of the vehicle of which it is a part, various degrees of hardening will be suitable to produce satisfactory resistance to failure in service. By way of illustration, Table 1 shows the practice in use just before the war by a number of builders of automotive vehicles. It will be seen that, even at the surface, the properties obtained in some instances, are not nearly as good as can be obtained with these steels. The columns showing percent of martensite show how far from full hardening is required. This chart illustrates beautifully how individual ideas and local circumstances determine the steel ultimately selected.

Table 1—Steering Knuckle Hardening Practice*

		Rockwell C Minimum Hardness Required Before Tempering			Minimum Required Hardness Expressed In Percent Martensite		
Critical Section	Steel Used	At Surface	At Half Radius	At Center	At Surface	At Half Radius	At Center
1.5-in. DIA.	X-3140 C-4140 8645	45	42	—	70	50	—
1.0-in. DIA.	4140	54	52	48	95	90	85
1-11/16-in. Thick	"	42	36	36	50	10	10
1-13/16-in.	5135	45	—	40	90	—	50
1-3/8-in. Thick	5135	45	—	39	90	—	50
1-5/8-in. Thick	8630	50	44	40	100	88	70
—	8647 4052	56.5	—	56	95	—	95
1-1/2-in. Thick	1340	44.5	—	29	70	—	0
1-3/4-in. Thick	9440	43	—	36	55	—	10
2.0-in. DIA.	1335	55	39	36	100	50	30

* Data Collected in 1944

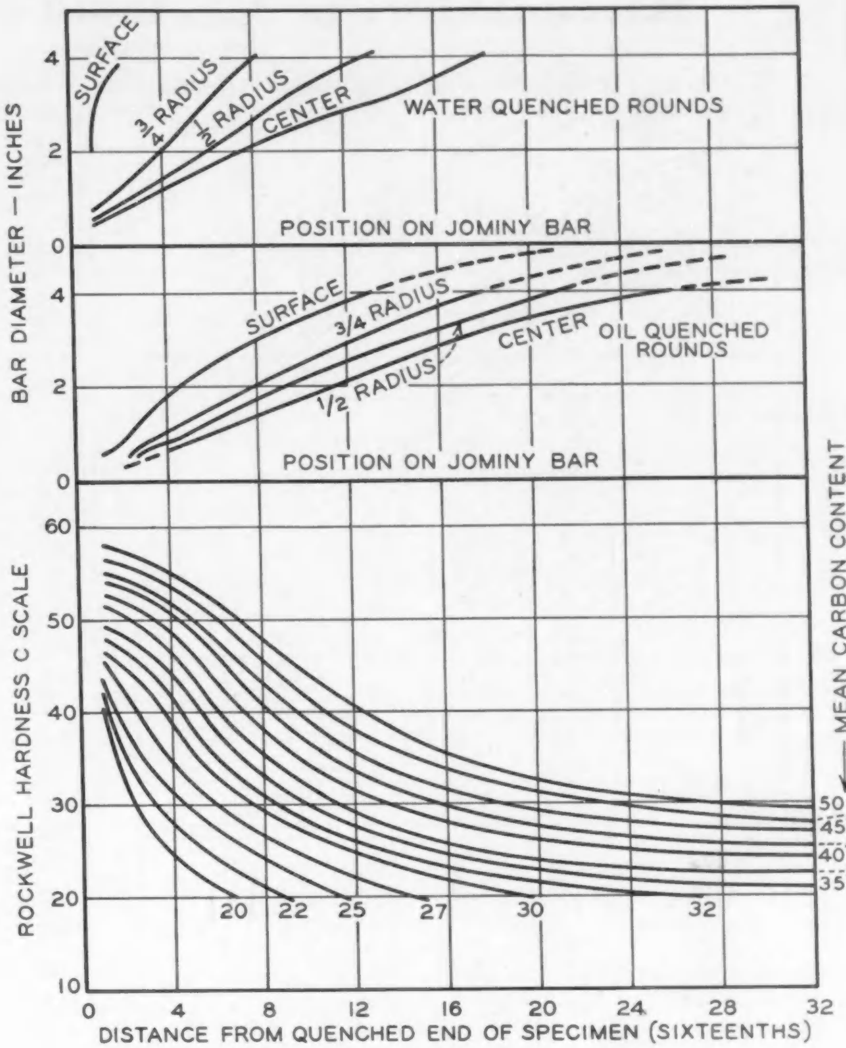


Fig. 24—Families of minimum H curves for the entire range of carbon contents in NE 8600H series steels

Out of eight manufacturers, no two are using the same steel; yet all the knuckles represented could be made of the same steel and heat-treated the same and they would all function satisfactorily.

The variety shown in this chart is the result of all the factors affecting cost at each of the plants represented. But whatever hardness is specified, either as the result of service tests or as the result of estimation of hardness required, the steel having sufficient hardenability to produce the desired hardness can be selected by the process of analysis illustrated by means of the steering knuckle chart (Fig. 17).

Matching Needs to Steels

As stated previously, when the desired minimum hardenability curve has been established, the next step is to compare with standard hardenability bands in the SAE-AISI tabulation to determine the steel having a minimum hardenability equal to the required curve. For ease of comparison we have assembled families of minimum H curves for the whole range of carbon contents in one type alloy steel, all on one page, as shown in Fig. 24. Included also at the top are curves for predicting hardness cross-section curves for any one of the H curves in the lower group.

Another Data Grouping

Another form of consolidated data is shown in Fig. 25, on p. 48. Here we have the predicted hardness in sections from ½-in. to 4-in. round, both for oil and for water quench and for minimum and for maximum hardenability of all 8500 steels, from 0.20% carbon up to 0.50% carbon content. These data plotted in the form of hardness cross-section curves for each of the 13 steels represented here would require 13 pages instead of one, as shown here. The steel finally selected will be one which will have a range of hardenability in a large number of heats, having a minimum sufficient to produce the desired hardness in the part in question.

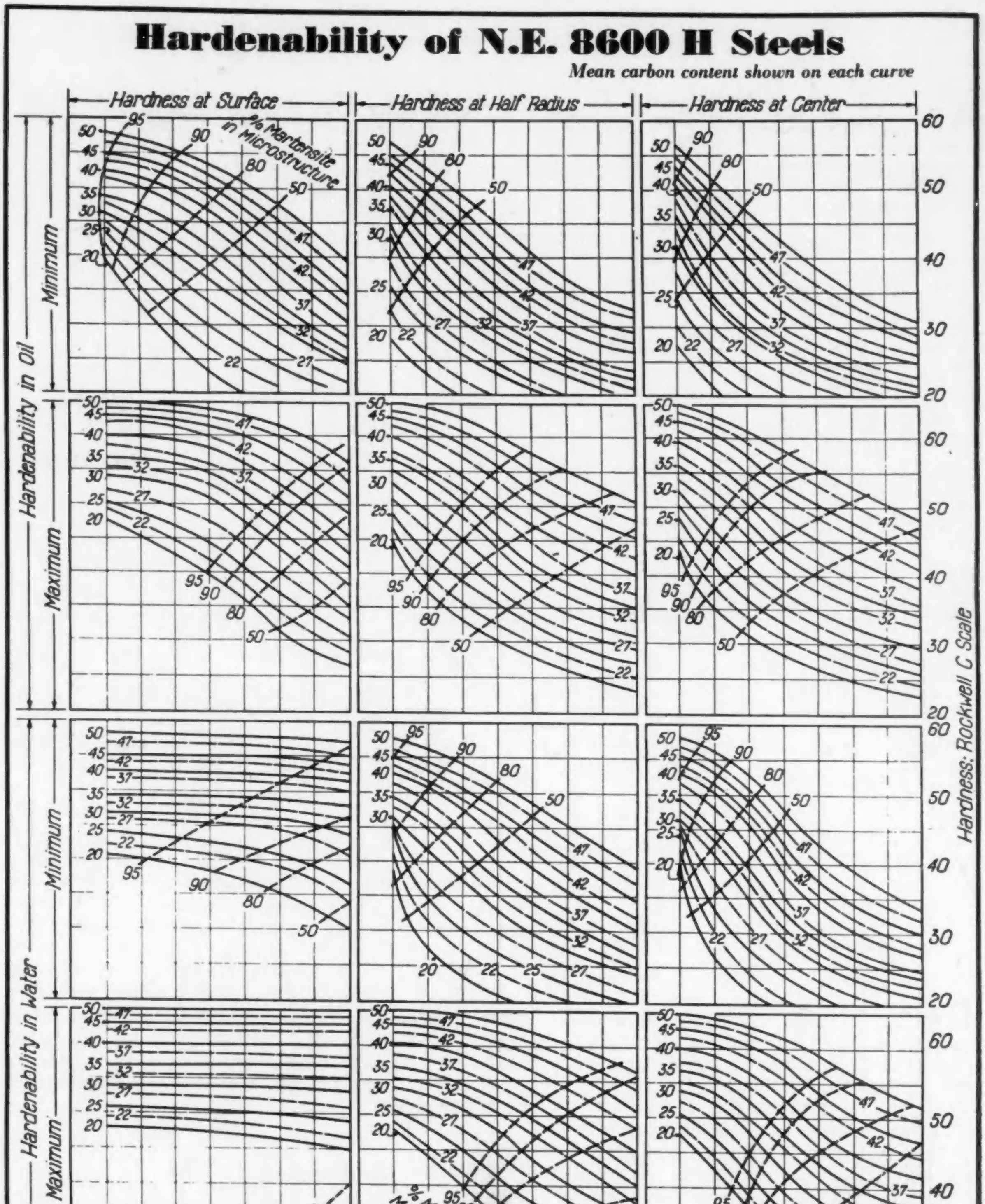


Fig. 25—Predicted hardness in section from ½ to 4-in. round both for oil and water quench, and for minimum and maximum hardenability of all 8600 steels

L. A. Water & Power Fleet's P. M. Program

EXCERPTS FROM PAPER* BY

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Automotive Repair Foreman

and Ellis W. Templin

Automotive Engineer,
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DURING and since the war years, the Los Angeles Power System of the Department of Water and Power has experienced a rise in transportation costs as have other fleet operators. To lower the costs on passenger car and truck repairs and operation, a thorough study of modern test instruments and procedures was made. As a result, engine dynamometers, a chassis dynamometer, and modern test instruments were purchased and suitable procedures and charts were worked out for the use of these instruments.

A program that is set on a strictly mileage or time basis for complete overhaul of engines, transmissions, power take-off, differentials, front ends, steering, and brakes will in numerous cases, due to the resulting discard of many slightly worn parts, increase the maintenance costs of most automotive fleets.

The Preventive Maintenance and Inspection Procedure schedule as outlined in the 1949 SAE Handbook, modified to fit Los Angeles Department of Water and Power practice, and shown in Table 1, has sufficient flexibility to maintain properly the Department's Power System automotive fleet of 1425 vehicles, including 838 trucks, 12 semitrailer tractors, 65 trailers, 496 passenger cars and 14 ten to twenty-five buses, safely and economically.

A Schedule, as in Table 1 applying to Department practice, is a daily maintenance service. However, in many cases where the vehicle is assigned to trips as well as local runs, this service may be partially repeated several times in a day. All vehicle tires are checked and aired at weekly intervals.

B Schedule maintenance service may vary with the area and conditions under which the vehicle is operating. The engine oil change is set for 2000 miles or every six months on passenger cars and light trucks; 1500 miles or every six months on trucks of 1½ tons capacity and up; and 1500 motor miles or every six months on light trucks equipped with motor-miles recorder. Large trucks equipped with winches, pumps, or compressors have their engine oil change set at 200 gal of fuel consumed or every six months.

All vehicles are inspected every 30 days or 1000 miles. The resulting shop order is referred to below. Dispatchers may request more frequent in-

* Paper "Modern Maintenance and Preventive Maintenance Procedures," was presented at SAE National West Coast Meeting, Portland, Oreg., Aug. 15, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Table 1—Los Angeles Department of Water and Power's Preventive Maintenance and Inspection Procedure Compared with SAE Recommended Practice*

Type of Vehicle	A	B Miles	C Miles	D Miles	E Miles
Passenger cars	Daily	1000 to 2000	4000 to 6000	30,000 to 50,000	60,000 to 100,000
Light trucks, stop & start service	do	(500 to 2000)	(3000 to 5000)	(20,000 to 30,000)	(40,000 to 60,000)
Light trucks, long haul service	do	1000 to 2000	4000 to 6000	40,000 to 60,000	80,000 to 120,000
Heavy trucks, stop & start service	do	(500 to 2000)	(3000 to 5000)	30,000 to 40,000	60,000 to 80,000
		1500 or Monthly			
Heavy trucks, long haul service	do	(1000 to 2000)	4000 to 6000	40,000 to 60,000	80,000 to 120,000
		1500 or Monthly			
Small buses, city service	do	1000 to 2000	3000 to 5000	30,000 to 40,000	60,000 to 80,000
Small buses all types of service					
Small buses, cross country	do	(1000 to 2000)	(4000 to 6000)	(40,000 to 60,000)	(80,000 to 120,000)
Large buses, city service	do	(1000 to 2000)	(4000 to 6000)	(40,000 to 60,000)	(80,000 to 120,000)
Large buses, cross country	do	(1000 to 2000)	(4000 to 6000)	(40,000 to 60,000)	(80,000 to 120,000)

* The Schedule is shown as given in the 1949 SAE Handbook, when in agreement with Department practice. Where items are in parentheses, they are not used by the Department. Where the department uses a different practice, that is indicated beneath the SAE practice.

DEPARTMENT OF WATER AND POWER
CITY OF LOS ANGELES

POWER GENERAL PLANT DIVISION
AUTO & CONST. EQUIP. REPAIR SHOP

CHASSIS DYNAMOMETER ENGINE ANALYSIS REPORT

Date .... Speedo In .... Car No. ....

Year .... Make .... Model ....

Oil Pressure : Cold .... Hot ....

Complaint or Procedure:

Fig. 1—This form is designed to record the results of vehicle tests on the chassis dynamometer before servicing, and those after recommended repairs and adjustments are made

DYNAMOMETER PERFORMANCE DATA

FULL LOAD

Motor Temperature 165° to 175°

MILES PER HOUR (Standard Speed)	ACTUAL HORSE POWER		CARBURETOR MIXTURE	
	Before Service	After Service	Before Service	After Service
25				
40				

CRUISING LOAD

Index Setting 17 H.P. at 40 M.P.H.

MILES PER HOUR		INTAKE MANIFOLD VACUUM		CARBURETOR MIXTURE	
Standard Speed	Car Speed Reading	Before Service	After Service	Before Service	After Service
40					
20					
Idle					

ACCELERATION

Speed Readings 10 to 50 M.P.H.

FUEL KNOCK: None : Light at M.P.H.: Heavy at M.P.H.

spection when car or truck has been on a desert or mountainous run before again assigning a vehicle to fleet service, regardless of miles run since last inspection. This procedure has eliminated many expensive road failures.

C Schedule includes many of B maintenance services together with a periodic engine oil analysis. The purpose of this analysis is to evaluate vehicle engine conditions in connection with the preventive maintenance inspection and thereby anticipate serious failures. Oil analysis is made at 4000 miles for passenger cars, and light trucks at 3000 miles or 3000 motor miles when equipped with a motor-miles recorder. For large trucks equipped with winches, pumps or compressors, engine oil analysis is made every 400 gal of fuel consumed.

The sludge condition in an engine is determined by the sludge index and solids test. Together they show whether or not sludge is forming, potential sludge is in circulation, or sludge is settling out. The engine oil analysis report recommends oil filters and engine breathers to be serviced when sludge or solids show high. This work is done if justified by actual observed conditions.

DIAGNOSIS

1. COMPRESSION PRESSURE:

1	2	3	4	5	6	7	8

2. PLUGS: Clean & Set ....
Replace ....

3. IGNITION CABLES: OK ....
Replace ....

4. DISTRIBUTOR: OK ....
Points .... Rotor ....
Plate .... Remarks ....
Cap ....
Bushings ....

5. CONDENSER: OK ....
Replace ....

6. COIL: OK ....
Replace ....

7. BATTERY CONDITION: OK ....
Connections ....

8. GENERATOR:
Overhaul ....
Remarks ....

9. REGULATOR: Voltage ....
Repair ....
Replace ....

10. STARTER:
Overhaul ....
Replace ....

11. FAN BELT: OK ....
Adjust .... Replace ....

12. CARBURETOR:
Replace ....
Recondition ....
Adjust ....
Vacuum ....
Remarks ....

13. CHOKE:
Clean and set ....
Adjust ....
Replace ....

14. FUEL PUMP:
Clean bowl ....
Flow and pressure ....
Replace ....

15. AIR CLEANER:
Clean ....
Replace ....

16. HEAT VALVE: OK ....
Free Up ....
Remarks ....

REMARKS:

....
....
....
....
....

Signature ....

The flushing of a crankcase is also recommended when engine crankcase deposits are high and this is done by using **SAE 10 oil** as a flushing medium. Crankcase oil temperature, not water temperature, should be maintained at a minimum of 160 F during flushing, with engine running at a fast idle, not to exceed 800 rpm, for 20 min.

With the engine oil analysis many other faulty engine conditions are exposed, such as piston rings sticking, dirt or sand in circulation, improper combustion, water and emulsions, bearing condition, and crankcase operating temperatures—whether low or too high. By making repairs to correct the faulty engine conditions, as recommended by the oil analysis report, before an engine failure has occurred, many costly major engine repairs have been eliminated in our fleet. This more than justifies the cost of the oil analysis.

Since the engine oil analysis is made at multiples of 3000 miles, 4000 miles, or 400 gal of fuel consumed, flexibility of the C, D, and E maintenance services is imperative; otherwise, duplication of operations

may occur, resulting in additional maintenance costs.

The inspector's shop order is filled out by the inspector, at the time of inspection, for the necessary repairs to the vehicle and sent to the coordinating dispatcher to be progressed through the appropriate repair shops. This procedure eliminates bottlenecks since the approval for repairs, service requirements of the vehicle, and dispatching of equipment to the shops are handled through the one office.

Tires are inspected every 3000 miles for tread or side wall defects, and excessive wear.

Tires of all sizes may be removed from a vehicle for recapping between the C and D Schedule, when the non-skid portion of the tread has worn down. Experience has proved that recapping has increased the life of a tire by 60% when capped at the proper time and correctly, resulting in a lowered tire mile cost.

Every 30 days batteries are serviced and tested, and cable insulation as well as the various terminal connections, battery container, and hold-down devices are inspected. Battery containers, battery hold-downs, and cable post connections corrode badly under certain types of vehicle operations. When corroding condition is present, regardless of time or mileage operated by the vehicle, battery containers and hold-downs are cleaned and painted with a suitable paint which resists the corroding action. Battery posts and cable terminals are cleaned and treated with vaseline after assembly, which retards the corrosion from again forming at these points.

All automotive batteries, other than those which were purchased with a new vehicle, are assembled by our battery maintenance shop. A recent study made of the length of life of several hundred batteries, turned over to salvage in the past year, showed an average of 32 2/5 months for those assembled by the battery shop as compared with much less service for those obtained with the new vehicle.

D Schedule is designed to fit the more extensive type of repairs (such as replacement of piston rings, adjustment or replacement of crankshaft bearings, complete engine tune up, repair of clutch, transmission, and differential bearings, or replacement of king pins and bushings) in which more time and equipment is required than in the previous schedules.

The mileage interval between C and D Schedules will also vary considerably due to type of service vehicle is operating in. It has proved advisable in our fleet to govern the repairs by the results of the preventive maintenance inspection and test, using the D Schedule as a guide as in the B and C Schedules.

The inspector's shop orders, written as the result of his tests and inspections, are also sent to the coordinating dispatcher's office, as in the C maintenance service, for transmittal through the transportation offices and to the automotive and construction equipment repair shops, if approved by the Superintendent of transportation.

Use of a chassis dynamometer is desirable for a more exacting diagnosis and thorough before-and-after repair checking, which simulates traffic and road grade conditions and indicates the vehicle's ability to perform on the road. As a substitute for a chassis dynamometer when the same is unavailable, a road test is made of the vehicle after the visual inspection using a vacuum gage and an air-fuel ratio analyzer, which gives the inspector a good picture of that particular vehicle condition.

Chassis Dynamometer Test

The chassis dynamometer at present is being used for passenger cars and light trucks for analyzing the vehicles and the setting of maximum horsepower output at the driving wheels.

During the period of engine heat normalization, the vehicle is installed on the dynamometer rollers, vacuum tube is attached to the intake manifold, and the air-fuel ratio analyzer tube is connected to the exhaust pipe. In conjunction with the chassis dynamometer, an engine analyzer is used and the various connections from the analyzer to the engine electrical circuits are also made. The chassis dynamometer engine analysis report heading is filled in with the equipment number, speedometer reading, oil presures, and so forth, all of which requires about 15 min. (See Fig. 1 for report form.)

The vehicle is then run at 20 and 40 mph, respectively, in direct gear with a fixed dynamometer index setting of 17 hp. Standard speed as shown on chassis dynamometer test report is actual driving wheel speed, recorded by the chassis dynamometer roller speedometer, and not the vehicle speedometer speed. This also gives a comparison test of vehicle speedometer calibration.

The manifold vacuum readings and carburetor mixtures are recorded under Cruising Load section in the "Before Service" column of the report. The vehicle is then run under full throttle conditions at 25 and 40 mph and maximum horsepower readings are recorded under Full Load section in the "Before Service" column.

With these two tests, the chassis dynamometer operator can properly analyze the condition of transmission bearings, drive lines, differentials, engine blowby, engine bearings, detonation, distributor, ignition timing, ignition coil, generator, generator regulator calibration, cooling system, exhaust system, carburetor, piston rings, valves, and the clutch.

The dynamometer load is then unloaded and the engine is accelerated several times from 10 to 50 mph and results recorded in the Acceleration section of chassis dynamometer test report.

The diagnosis section of the report, (see right-hand form in Fig. 1) is then filled in by the chassis dynamometer operator, together with a shop work order for the necessary repairs to the engine and the various units. Engines that have 10 psi or more variation in compression pressure, cannot be tuned for maximum horsepower efficiency or satisfactory service for fleet operation.

After the recommended repairs are made, the vehicle is again run on the chassis dynamometer and final adjustments are made. The operator then fills in the "After Service" column of the analysis report for record purposes. The vehicle is then ready for fleet service.

The entire procedure of before-and-after service runs, together with a complete engine tune up, requires on the average 4 hr.

DEPARTMENT OF WATER & POWER
CITY OF LOS ANGELES

POWER GENERAL PLANT DIVISION
AUTO & CONST. EQUIP. REPAIR SHOP

MOTOR ANALYZING REPORT

Date .... Speedo .... Car No. ....

Make .... Type .... Engine ....

Oil Pressure: Cold .... lbs. Hot .... lbs. S.A.E. 10 oil .... lbs.

Compression pressure before and after solvent treatment:

	1	2	3	4	5	6	7	8
Before								
After								

Fig. 2—The motor analyzing report is filled in after minor repairs and adjustments are made, with suggestions noted on it for major repairs

Valve clearances before adjustment: After adjustment:

	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
Intake																
Exhaust																

Vacuum .... in. to .... in. Sludge ....

Spark plugs: Make .... Type ....

Spark plugs installed

Primary Test

Condition of generator ....Ampere output ....Volt output ....

Secondary Test ....Coil Test ....Condenser Test ....

Condition of distributor ....Ignition Timing ....

Condition of starter ....Battery Test ....

Condition of cooling system .... Fuel pump pressure .... lbs.

Condition of carburetor .... Fuel pump vacuum .... ins.

Air cleaner: Type ....Condition ....

Exhaust analyzer readings:

Idle Ratio .... in. of vacuum .... 35 MPH Ratio .... in. of vacuum ....

20 MPH Ratio .... in. of vacuum .... 40 MPH Ratio .... in. of vacuum ....

25 MPH Ratio .... in. of vacuum .... 45 MPH Ratio .... in. of vacuum ....

30 MPH Ratio .... in. of vacuum .... 50 MPH Ratio .... in. of vacuum ....

REMARKS:

Prior to the use of the chassis dynamometer, an average of an additional 2 hr per vehicle was required for the analysis and road adjustments; in many cases, due to city traffic conditions, inadequate runs were made for the setting of the various engine units, resulting in frequent vehicle failures.

A complete engine tune up is often found necessary between the C and D maintenance service, which takes into consideration the condition and adjustments of all the related units of an engine. It will not only give a smoother engine performance, but has eliminated many costly repairs to clutches, transmissions, drive lines, and differentials and has lengthened piston-ring life.

With the tune-up procedure, a good tune-up solvent is used which frees the valve action and piston rings, develops an additional 2 to 5 hp at the driving wheels, and has, therefore, become a standard practice with many fleet operators as well as ourselves.

A complete tune-up procedure, as used by our automotive and construction equipment repair shops, consists of an analysis, solvent treatment, testing and adjusting the engine and its various units. This procedure is also used at the district

Dept. of Water and Power
City of Los Angeles

General Plant Division
Auto and Const. Equip. Repair Shops

Engine Dynamometer Break-in Report

Date ______ Make ______ Job No. ______ P. No. ______

Engine Make ______ Model ______ Engine No. ______

Oil Pressure: Cold ______ lbs. Hot ______ lbs. Idle Hot ______ lbs.

Spark Plugs: Make ______ Type ______ Ignition Condition ______

Fuel Pump: Pressure ______ lbs. Vacuum ______ In. Blowby ______ Ign. Timing ______

Generator Output: Volts ______ Amps. ______ Battery Test ______ Starter Test ______

Primary Test ______ Coil Test ______ Condenser Test ______ Secondary Test ______ %

Carburetor Condition ______ Air Cleaners: Carb. ______ Eng. ______

Compression Pressure after break-in.

1	2	3	4	5	6	7	8

	Valve Tappet Clearances								Main Bearing Temperatures							
	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
Intake																
Exhaust																

Oper.	R.P.M.	Time Run	Vacuum	Oil Pres.	Oil Temp.	Engine Water Temp.	H.P. Indicator	Fuel Analysis
1								
2								
3								
4								
5								
6								

Governor Setting ______ R.P.M. Oil Regulator Setting ______ lbs. Date Started ______

Speedo Reading: Out ______ In ______ Date Finished ______

Remarks:

______
Test Made By

Fig. 3—This form is used to report results of breaking in an overhauled engine on the engine dynamometer

shops where a chassis dynamometer is not available. However, a different record chart is used entitled "Motor Analyzing Report." See Fig. 2.

At the district shops, during the period of engine heat normalization, the motor analyzing report heading is filled in. Individual cylinder compression is then taken with a compression tester before and after solvent treatment of the engine, because valve heads may be burned, piston rings scored, or frozen in the piston grooves. When these conditions exist or compression pressures vary more than 10 psi per cylinder, the tune up of an engine is not recommended until major repairs have been completed.

The tune-up man fills in the motor analyzing report as the tests and minor repairs and adjustments are completed. When major repairs are needed, such as reconditioning of valves or replacement of piston rings and crankshaft bearings, a shop work order is written and a notation made on the motor analyzing report of the recommended repairs, under "Remarks."

New pieces of automotive equipment, as received from the various dealers, many times do not con-

form to the Power System's standards, since they are often found with loose wire connections, faulty carburetors, incorrect distributor cam angle settings and ignition timing, sticking valves, loose engine mounting bolts, defective water thermostats, generator regulators not calibrated, and defective braking system. With the analyzation and tune-up procedure, these conditions are detected and either corrected by the dealer or our shops before the new vehicle is assigned to fleet service. This has proved to be a definite factor in repair cost savings.

E Schedule maintenance service covers, in addition to the C and D services, the more extensive vehicle repairs, such as the major overhaul of the vehicle's various units. Type of vehicle, operating conditions, and amount of repair done in B, C and D services are also the primary factors affecting the E maintenance service.

The analysis and diagnosis of vehicles for the more extensive E service repairs depend not merely on the miles vehicle has run, but on the extent of repairs in the C and D maintenance services. It is also practically impossible to make an intelligent inspection of many vehicle unit parts, axles, spindle bodies, and drive shafts until they have been disassembled and thoroughly cleaned.

Preventing Future Breakdown

Many axle shafts, drive shafts, spindle bodies, transmission cluster gears, and steering arms have inherent discontinuities which do not necessarily mean the part is defective; they are often replaced with a potential life or mileage still in them for the particular type of service the vehicle performs, which results in apparently unnecessary maintenance costs. There are times when this replacement policy is justified, when the risk of an early failure would result in a still larger expenditure.

Engines in our fleet, when approved for repair with the E maintenance service, are completely dismantled, thoroughly cleaned, and all parts inspected for defects. Cylinders are rebored to a standard oversize and new pistons are ground to the correct clearances for that engine. Valves are reconditioned with precision tools. Crankshaft journals are reground when necessary and new crankshaft bearings are line bored to assure the correct crankshaft alignment. Bell housing is checked with a dial indicator for alignment with crankshaft flange.

Cylinder block and head faces are reconditioned when necessary with a surface grinder, eliminating leaking head gaskets and uneven compression.

All overhauled engines are now run in and tested with the engine dynamometer before re-installing in a car or truck chassis. Incorporated in the engine dynamometers are oil pressure gage, water temperature gage, vacuum gage, tachometer and horsepower indicator. Distributor cam angle is calibrated with a Distribuscope, and the ignition timing is set with a timing light and a vacuum gage to the highest point of engine efficiency, as determined by the horsepower indicator, without detonation. The carburetor mixture is tested with an air-fuel ratio analyzer and the fuel pump for leaking valves, defective diaphragms and correct pressure, with a vacuum and pressure gage.

Wiring circuits are tested with an electric trouble shooter, and repaired if necessary, before the engine is re-installed in a vehicle chassis.

Overhauled engines which are run in with the dynamometer are being delivered to the various district shops and exchanged from other engines of the same make and model for installation in chassis. This eliminates further testing of engines in the district shop.

The information requested in the engine dynamometer break-in report is filled in as the various tests of the engine units are made. See Fig. 3.

Defects Caught

The engine dynamometer, with the break-in procedure, has detected collapsed pistons, faulty piston rings, defective oil pump check valve, defective water thermostats, bearings operating with too much crankshaft clearance, and inadequate oil volume to accessory shaft bearings. The defective engine conditions are corrected before the engine is re-installed in a car or truck chassis, and this has resulted in an appreciable savings in labor cost.

The section of engine dynamometer break-in report, showing operations one through six, is filled in at 1-hr intervals, or at the time the engine loosens up to the point where artificial load can be increased safely. The ignition timing final setting, visual blowby, generator output, and secondary test are made at the time of full throttle run for maximum horsepower.

On completion of the break-in run, break-in oil is removed and replaced with new oil of the correct viscosity for that engine. Compression pressures are taken, and then while the engine is running hot, valve tappet adjustments are checked as well as the oil regulator and governor settings. If necessary, new spark plugs are installed which have the correct heat range for that particular engine, according to the type of service it is expected to perform.

The time consumed for installing an engine on the dynamometer, filling in the required data on the engine dynamometer break-in report, and completing the break-in run, averages 8 hr for a passenger car and 10 hr for a heavy truck engine. Prior to this procedure, the average break-in period and road test required three 8-hr days per vehicle.

Major brake overhauls include the cleaning, inspecting, and reconditioning of drums, shoes, cross shafts, linkage pins and clevices, hydraulic cylinders, air chambers, boosters and other related parts as well as the relining, centralizing, and grinding of of lined shoes to the brake drum diameter. When necessary and advisable, the drums are reground to a true cylindrical and smooth surface.

Preventive maintenance procedures do not eliminate the need of a maintenance procedure schedule for the economical operation of a large fleet. Proper dispatching, vehicle driving, cooperation with the maintenance shops, lubrication and polishing, as well as tire and battery service will probably be the most important factors in holding the operating and maintenance cost curve to a minimum. More frequent testing and minor repairing will generally show an uptrend of the maintenance cost curve; but they are imperative when a vehicle fleet is being maintained in such a manner as to be able to meet any emergency that may arise in the operation of a large power system such as ours.

ALUMINUM-IRON DRUM VEHICLE

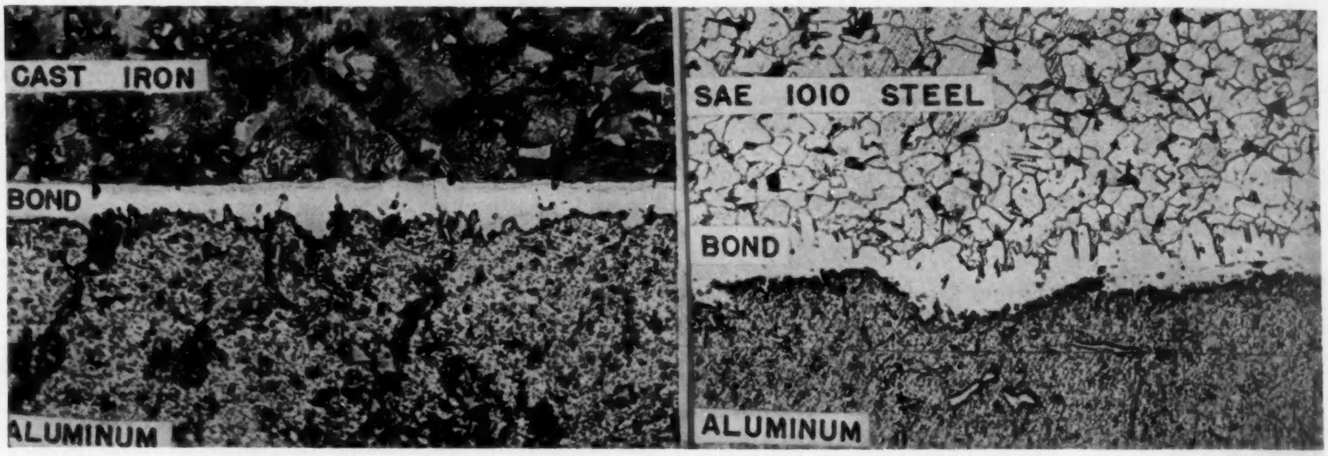

Fig. 1—Microphotographs (250 diameters) of the molecular bond between aluminum and ferrous materials

(This paper will be printed in full in SAE Quarterly Transactions)

AN automotive aluminum-cast-iron brake drum, stemming from development of a bimetallic cylinder barrel for aircraft engines, holds much promise because of its heat-dissipating properties. Essentially a heat exchanger, this bimetallic brake dissipates large amounts of energy without exceeding a temperature at which linings lose their effectiveness.

By substituting this heat exchanger for a heat reservoir, it may be possible to strike a balance between heat-in versus heat-out so that the brake could be used steadily on down grades, or for steady checking at high speed in heavy traffic, without getting so hot that the lining fades and the brake becomes ineffective.

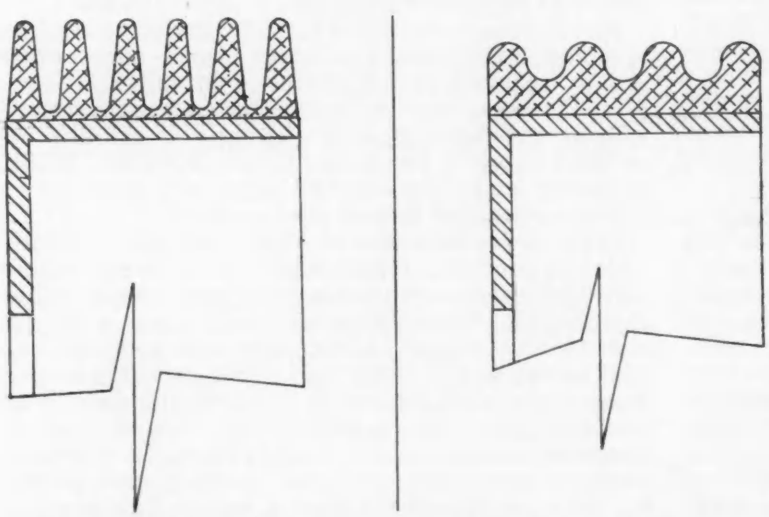

Fig. 2—Bimetallic brake drum cross-section at left withstood high temperature testing while the one at right did not

War work pointed up the greater heat-dissipating capacity of bimetallic cylinder barrels for aircraft engines, as compared with steel fin barrels. Test runs established that a molecular bond is formed between the aluminum and ferrous members (see Fig. 1), as that between copper and steel, zinc and steel, and other nonferrous metals and steel.

A brake drum is not too distant a concept from a cylinder barrel. In both cases a ferrous liner must be used, and for the same reasons—strength and resistance to abrasion. Both are subjected to high heat and both are designed to dissipate that heat by an extended surface design. But many factors other than promise of usefulness and a method for applying a metal of high heat con-

BOOSTS BRAKE PERFORMANCE

BASED ON PAPER* BY

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ductivity must be considered in design of a bimetallic brake drum. The test and development program of the bimetallic brake drum uncovered its potential advantages as well as its problems.

Initial testing was done in a heat-treating oven, rather than on a vehicle or dynamometer, since the thermal stress problem had to be satisfied before subjecting the construction to mechanical stresses. Coefficient of expansion of aluminum is about 13 x 10^{-6} in. per in. per deg F, whereas that of cast iron is about 5.6 x 10^{-6}. This could lead to a separation of the two materials at the interface, unless the design were such that the bond strength could restrain the aluminum and keep the construction intact.

Fig. 2 shows two constructions. The design at left successfully withstood thermal testing, while the one at right failed by bond separation at about 450 F. Examination of these shows intuitively that the design motivation is to reduce the aluminum cross section until it is less strong than the bond; therefore, it will be held by the bond and will assume the dimensions dictated by the iron.

After developing a design that combined good heat transfer with ability to withstand all thermal testing, a set of drums of passenger car size was made and tested on a brake drum dynamometer. The results, shown in Figs. 3 and 4, were encouraging.

One aspect of the construction considered in this test was the deflection under operating conditions. Plot of the deflection of a bimetallic brake drum versus a standard drum is shown in Fig. 5.

Next test was the first on an actual vehicle—a 1500-cc midget racing car, of latest body and chassis design, powered by an Offenhouser engine.

Racing people looked upon the severe brake drum problem, posed by this midget car racing on flat one-fifth mile asphalt-paved tracks, as unsolvable. One turn of the track is made every 15 sec, with two brake applications per lap. So great were demands of such driving on the brake that the driver usually

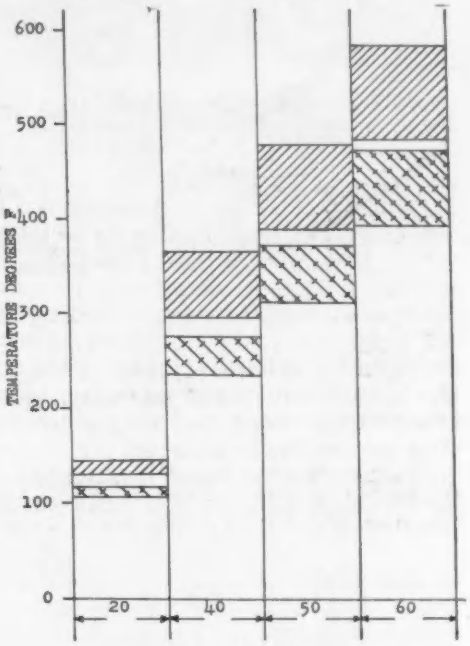


Fig. 3—Stable temperature ranges for repeated stops at various speeds. Test drum is 11 x 2 in., designed for passenger car service

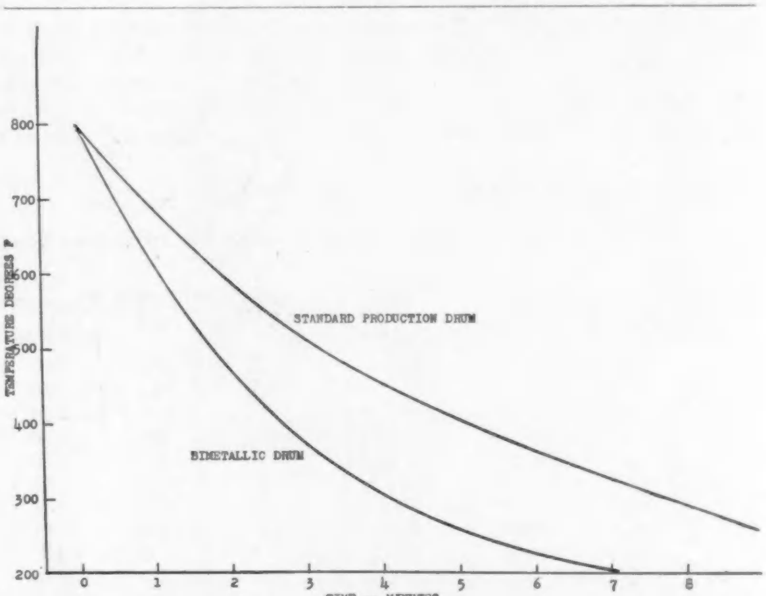


Fig. 4—Cooling curves from 800 to 200 F for bimetallic and standard cast-iron brake drums for passenger cars, 11 x 2½ in. Tests were run with fans operating on drums

* Paper "Design and Development Considerations of a Bimetallic Brake Drum," was presented at SAE National West Coast Meeting, Portland, Oreg., Aug. 16, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

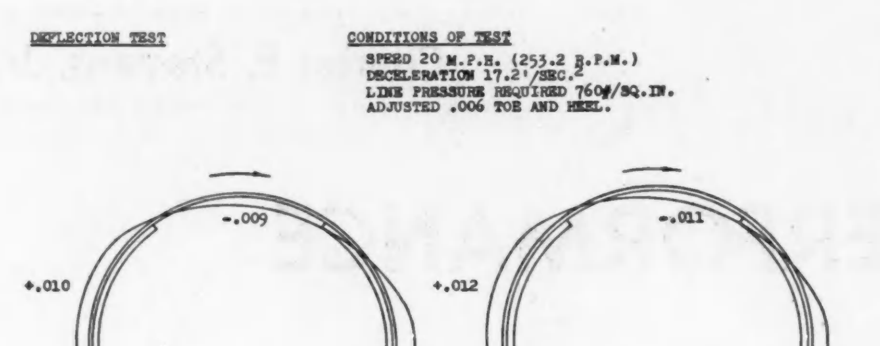


Fig. 5—Deflection of bimetallic and cast-iron brake drums under identical conditions

changed his driving style after the fifth lap because the brakes had faded so badly. Brakes had to be relined every week because the leading inch of each shoe was so badly charred.

After installing bonded bimetallic brakes on the front end of this car, Fig. 6, the driver said that for the first time in his racing career he could get the same brake effect throughout a race, no matter what its length. He won five of his next six starts and the linings lasted throughout the season.

Fact to be emphasized here is that this application is ideal for a heat exchanger type of brake. As the construction in Fig. 6 shows, the drums are in the air stream. Additionally, the operation keeps the brake working in a minimum air stream of 40 mph and a maximum air stream of 80 mph.

Third step in the test program consisted of making a drum for a large-capacity commercial vehicle, such as an intercity bus or heavy truck. First one made was a 16.5 x 5-in. drum, of the cross-section shown in Fig. 7. This drum was given 2750 stops at varying energy levels on a dynamometer and showed excellent thermal dissipation. Table 1 shows typical stabilization test figures compared with these of a conventional cast-iron drum. At the end of the test, the bimetallic drum was still performing well, although bond loosening seemed to be indicated.

Next, two such drums were placed in service, mounted on the rear wheels of a medium-sized intracity bus, and have been satisfactory for a year. The drums were inspected at 26,000 miles and did not need refinishing at that time. There has been no brake squeal with these drums—unique in this operation.

Subsequent examination of the drum tested on the dynamometer did show that the bond failed along the outer or open end. This clearly indicates that the combined stress created by the superimposing of mechanical stress, caused by the deflection, on the thermal stress was too great.

A design was tested with a modest flange on the outer end to combat this tendency. After complet-

Table 1—Stabilization Temperatures of Several Bimetallic and Cast-Iron Brake Drums*

Test No.	Material	Size	Speed M.P.H.	Deceleration Ft/sec²	Stabilization Temperatures—deg F		Cycle Time	Wheel Load (lb)	Type of Cooling
					Drum Max.	Lining Max.			
1	Al. Bimetallic	16½ x 5	20	12.5	210	...	3'00"	9000	Suction
2	Al. Bimetallic	16½ x 5	30	12.0	460	...	3'00"	9000	Suction
3	Al. Bimetallic	16½ x 5	40	11.0	510	...	3'00"	9000	Suction
4	Al. Bimetallic	16½ x 7	30	8.0	235	290	1'25"	10,000	Fan
5	Cast Iron	16½ x 7	30	8.0	340	300	1'25"	9000	Fan
6	Al. Bimetallic	16½ x 7	40	10.0	385	425	1'40"	10,000	Fan
7	Cast Iron	16½ x 5	40	11.0	655	...	1'30"	6000	Suction
8	Al. Bimetallic	16½ x 7	50	10.0	505	530	2'10"	10,000	Fan
9	Cast Iron	16½ x 5	50	11.0	600	...	3'00"	6000	Suction
10	Al. Bimetallic	16½ x 7	50	10.0	680	690	1'32"	10,000	Fan
11	Al. Bimetallic	16½ x 7	55	11.3	730	720	1'40"	10,000	Fan

* Each test consisted of at least 100 stops.

ing all the testing on this drum, there was no indication of any failure; addition of the outer bead had eliminated that.

But one thing that had not been considered was the increase in deflection due to the decrease in the modulus of both cast iron and aluminum with temperature. This made itself felt in the last of a series of five fade stops without cooling, in which, at a 640 F temperature, the actuating piston reached the end of its possible travel with a resulting low deceleration value. The current drum being readied for tests has the cross-section shown in Fig. 8 and gives computed deflections comparable with that of the best cast iron at temperatures up to 650 F.

Results of all this testing lead to one conclusion: Where cooling air is available, the bimetallic drum will out-perform the standard all cast-iron construction. When cooling air is not available, then the use of this drum is of dubious advantage, except in the case of squeal elimination.

Cooling curves of a bimetallic versus a standard drum, made on a test stand, but with the wheels mounted in position, show the great advantage of cooling air. See Fig. 9. The curves also show the relatively small advantage in providing cooling air to the standard drum. The property of fast recovery between stops is extremely valuable, since it permits the bimetallic drum to start its braking at a lower temperature level and not build up to impossible temperatures with continued use.

A severe test recently run best illustrates this property. These data were gathered during a fade test consisting of stops every 45 sec at a torque level that gave a deceleration of 18 ft per sec^2 initially. Note from the data in Table 2 that the deceleration produced by the standard drum has faded in effectiveness to less than one-third its value in six stops, while the bimetallic drum maintained a deceleration of greater than one-half the initial value at the end of 17 stops. And in the latter case, the temperature also has been stabilized.

Both drum temperatures were taken at a location about 1/16 in. from the friction surface. The bulk of the cast iron was so sluggish in its response to temperature that by the sixth stop, the temperature of the cast-iron drum rubbing surface probably exceeded 1000 F. Friction surface temperature of the

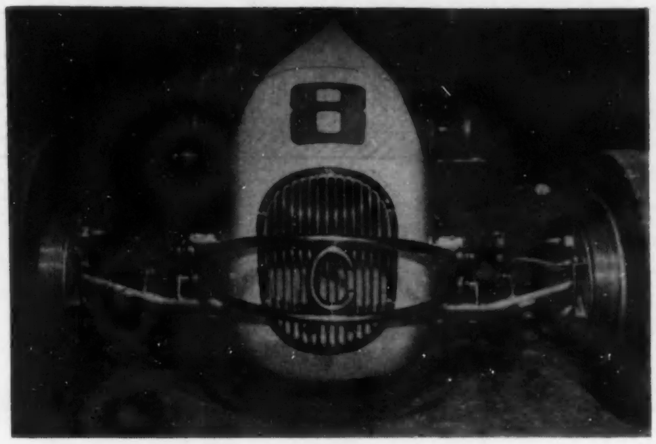
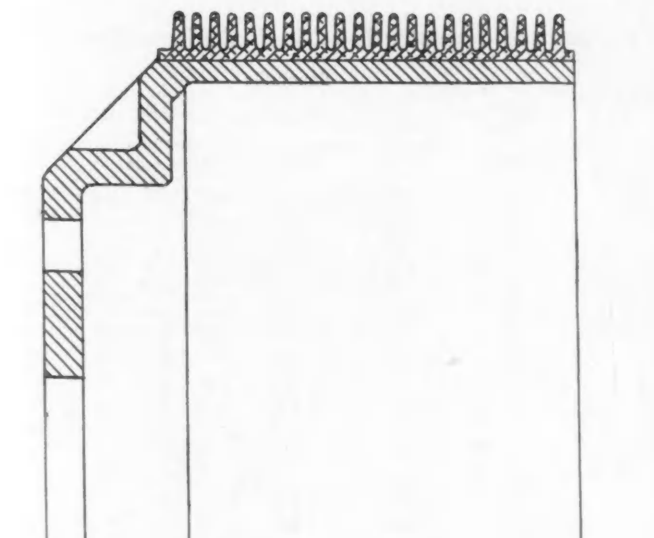
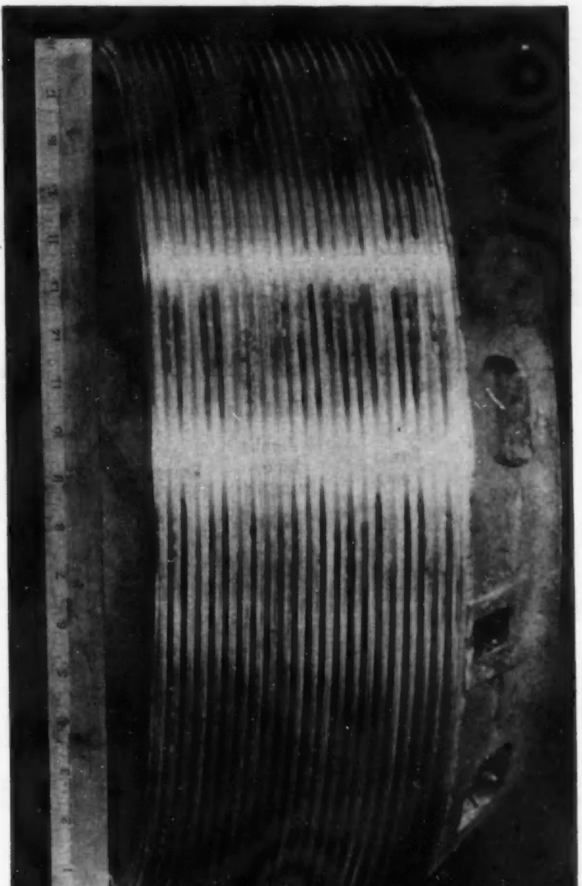


Fig. 6—Bimetallic brake drums installed on this midget racing car increased lining life considerably and prevented brake fade after several brake applications

Fig. 7—First commercial bimetallic brake drum, 16½ x 5 in.





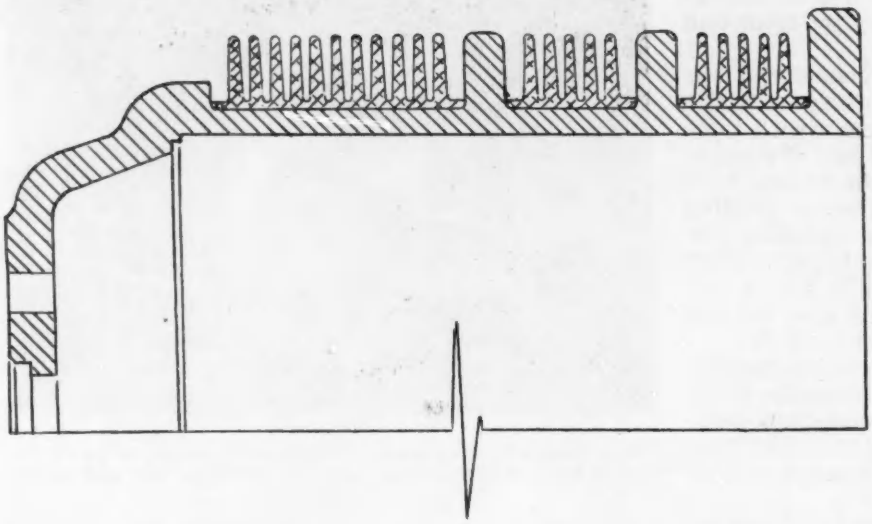


Fig. 8—Cross-section of the current bimetallic brake drum, designed to conform to the deflection pattern of the best cast-iron drums

bimetallic drum could not have been much above the eventual stabilization temperature of 840 F. Cooling air during these tests had a speed of 25 mph.

Another recently-run test shows the same type of result as well as the advantage of cooling air on the bimetallic brake drum. See Fig. 10.

Demonstrated throughout all the tests is the fact that bimetallic drums perform better than reasonable expectations without cooling air. The average bimetallic drum weighs less than two-thirds as much as the all cast-iron drum it replaces. In the case of the drums used for the tests reported in Fig. 10, the cast-iron drum weighed 137 lb, the bimetallic one, 83 lb. This extra weight represents added specific heat which should be of initial advantage.

None of the cooling used in these tests has been of the high-speed high-pressure type, as is used in cooling aircraft engines. Some consisted of only inexpensive house fans; other installations used fractional horsepower blowers, often located some distance from the drum. In all cases it is felt that road conditions were simulated.

Perhaps this is optimistic, although it is reasonable that, in operating a vehicle, the additional air could be redirected by deflecting scoops to impinge on the fins to give results comparable to those on the dynamometer.

Full cooling potential of the bimetallic drum can be derived with high-speed power cooling. If there are such severe braking problems that make economically feasible the installation of a fractional horsepower blower, then the bimetallic brake drum (on the basis of scanty data gathered to date) offers a potential in performance formerly thought impossible. The cast-iron drum presents no such potential.

At present the bimetallic brake drum for passenger cars offers more of an improvement than any other drum-type brake. Brakes of this size pose no stress problems that require further testing to prove the design. Extensive dynamometer test-

Table 2—Fade Test Comparison Between Cast-Iron and Bimetallic 11 x 2-in. Brake Drums*

Standard Cast Iron					Bimetallic				
No. of Stops	Drum Temp.	Hydraulic Line Pressure	Deceleration: Stopping Time (sec)	Deceleration: Ft/sec²	No. of Stops	Drum Temp.	Hydraulic Line Pressure	Deceleration: Stopping Time (sec)	Deceleration: Ft/sec²
1	420	790	5.0	17.6	1	340	650	5.0	17.6
2	500	790	7.1	12.4	2	420	650	6.0	14.6
3	560	790	9.1	9.7	3	480	650	6.9	12.8
4	600	790	12.1	7.5	4	570	650	7.4	11.9
5	650	790	14.9	5.9	5	600	650	8.0	11.0
6	660	790	16.5	5.3	6	670	650	8.5	10.3
					7	665	650	9.0	9.8
					8	710	650	9.2	9.6
					9	710	650	9.5	9.3
					10	760	650	9.8	9.0
					11	800	650	9.8	9.0
					12	780	650	9.8	9.0
					13	810	650	10.0	8.8
					14	780	650	10.0	8.8
					15	830	650	9.9	8.9
					16	850	650	9.7	9.1
					17	840	650	9.8	9.0

* Tests were made at 60 mph, with 45-sec intervals between stops. Blower air speed was 25 mph.

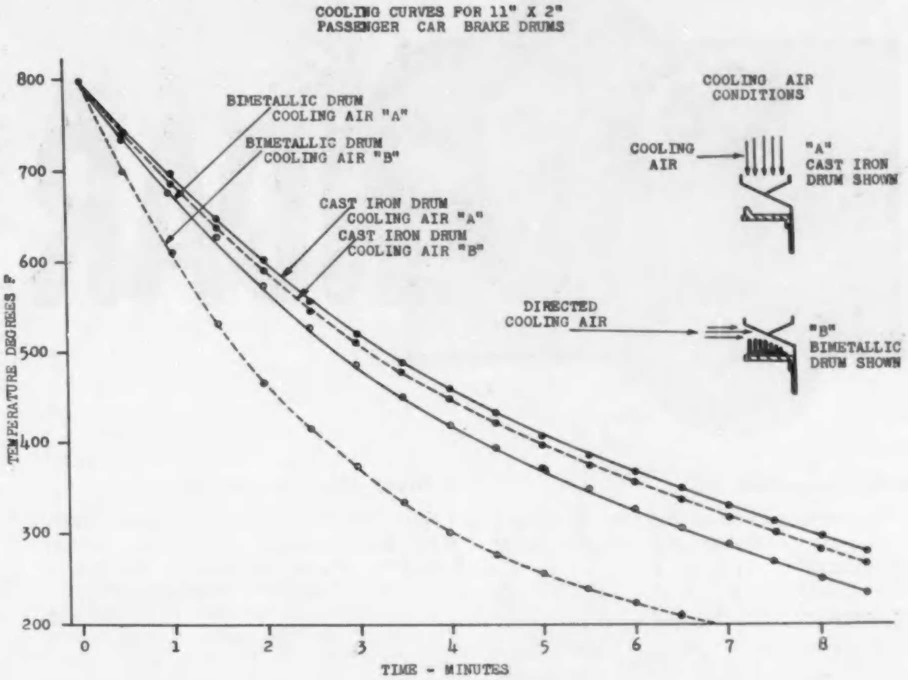


Fig. 9—These cooling curves illustrate the gains from properly directing cooling air to the bimetallic brake drum

ing plus the year of service on the racing car have failed to produce a stress failure.

Increasing use of fluid couplings somewhat reduces the engine's braking effect at low speeds and places more responsibility on the brakes. Adding deflecting shields or scoops is not difficult and would not disturb current styling. While it might be better to redesign the wheel-brake-spindle assembly to place the brake drum in the air stream, this probably is not necessary for current needs.

Bimetallic brake drums of this size could be permanent-mold cast in large quantities at a cost not much higher than that of present high-quality drums. The day is near when large-capacity commercial drums should be available. The drum construction shown in Fig. 8 will soon be made and several of these should be thoroughly dynamometer and field tested. The finned configuration on this large-size drum has been successfully sand cast; thus it should not command too high a premium since it will not be machined, as were those of all the test drums.

In addition to its thermal properties, other considerations seem to favor the bimetallic brake drum. For example, the greatly reduced thickness of cast iron probably will reduce the tendency to check or craze. This slight cross-section, backed up by the high heat dissipation of a good fin design, greatly reduces the temperature gradient across the cast iron. If minute upsetting during high energy applications causes checking, as is theoretically likely, then the cast-iron portion—being more nearly uniform in temperature—should reduce this tendency.

There is also evidence that bimetallic brake drums do not normally squeal. This may be a function of the natural resonance of an all cast-iron drum being broken up or dampened by another metal of different tonal characteristics. While there is yet insufficient field testing to verify positively these last two characteristics, testing to date does show promise that one or both will be realized to some degree.

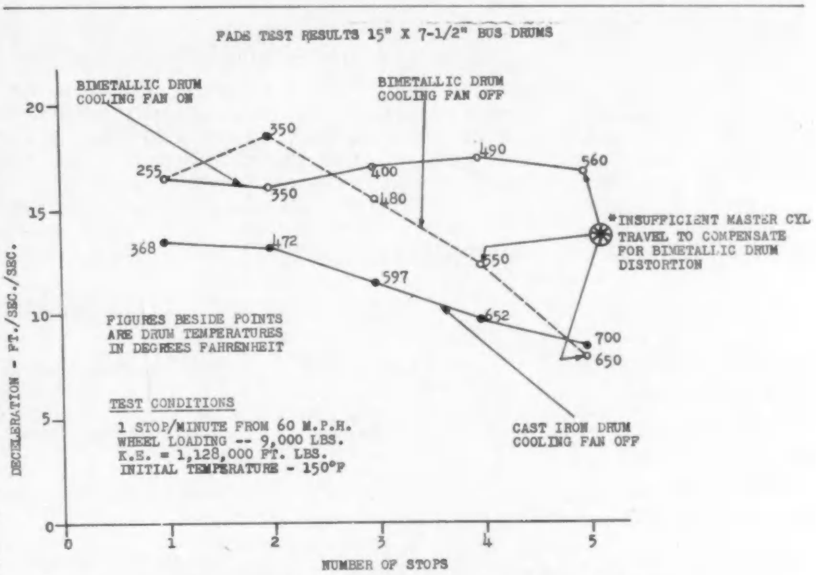


Fig. 10—Fade test results also point up the advantage of cooling air for bimetallic brakes

CALENDAR

Baltimore—Nov. 10

Engineer's Club of Baltimore; dinner 7:00 p.m. Speaker and subject to be announced.

Central Illinois—Nov. 21

Hotel Jefferson, Peoria, Ill.; dinner 6:30 p.m. Meeting (for members); 7:45 p.m. Choosing Proper Testing Environment—Frank A. Grooss, staff engineer, Caterpillar Tractor Co., Peoria. Experimental Stress Analysis —Harry W. Fall, supervisor, research department, Caterpillar Tractor Co., Peoria. Why Service Department?—Theodore M. Fahnestock, mechanical engineer, service engineering division, Caterpillar Tractor Co., Peoria. After dinner speaker to be announced.

Cleveland—Nov. 14

N.A.C.A. Laboratory; dinner 6:30 p.m. Gas Turbine Design for Producibility—A. T. Colwell, vice-president, Thompson Products, Inc., Cleveland, Ohio. Speaker-sponsor: E. E. Bisson.

Dayton—Nov. 10

Buffalo-Springfield Roller Co., Springfield, Ohio; dinner 7:00 p.m. Antique Automobile Show. Road Building and Its Relation to the Automotive Industry—Major E. E. Greiner, Carl F. Greiner, John F. Harrison and Murray D. Shaffer.

Detroit—Nov. 14 and 28

Nov. 14—Large Auditorium, Rackham Educational Memorial; dinner 6:30 p.m. Panel meeting on An Evaluation of Criticisms of Modern Automotive Design. Moderator: F. S. Spring, director of styling, Hudson Motor Car Co. Speakers: P. C. Ackerman, chief engineer in charge of laboratories, Chrysler Corp.; H. K. Gandelot, safety engineer, car design, General Motors Corp.; and John Oswald, executive engineer, styling and body engineering, Ford Motor Co. Dinner speaker: Gibson Bradfield, president, McClelland-Kennard Co. Subject: Boat Racing and Race Boats.

Nov. 28—Champion Spark Plug Co., Ceramic Division, 8525 Butler, Hamtramck, Mich. Field trip through plant 2:00 p.m.

Indiana—Nov. 10

Hotel Antlers, Indianapolis, Ind.; dinner 7:00 p.m. Recent Developments in Engines, Parts and Fuel—A. T. Colwell, vice-president, Thompson Products, Inc., Cleveland, Ohio.

Kansas City—Nov. 14 and Dec. 13

Nov. 14—Plaza Cafeteria, 414 Alameda Rd., 'On The Country Club Plaza'; dinner 7:00 p.m. My Friend, the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., and president, SAE. Social half hour 6:30 p.m. Meeting 8:00 p.m.

Dec. 13—Plaza Cafeteria, 414 Alameda Rd., 'On The Country Club Plaza'; dinner 7:00 p.m. Automotive Brake Problems—Paul J. Reese, manager, Bus & Tractor Engineering, Wagner Electric Corp. Social half hour 6:30 p.m. Meeting 8:00 p.m.

Metropolitan—Nov. 17

Hotel Statler, Keystone Room; meeting 7:45 p.m. Greases—George Link, Shell Oil Co. Discussors: Representatives of Swift Co. and Texas Oil Co.

Mid-Continent—Nov. 18

Hotel Biltmore, Oklahoma City, Okla.; dinner 6:30 p.m. My Friend, the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., and president, SAE.

New England—Dec. 6

Graduate House, Massachusetts Institute of Technology; dinner 6:30 p.m. Speaker and subject to be announced.

Northern California—Nov. 21

John's Cafe, San Leandro; dinner 6:30 p.m. Meeting 7:30 p.m. Field trip through the San Leandro plant of the Caterpillar Tractor Co. 2:00 p.m. Meeting will feature a Caterpillar Tractor Co. film.

Philadelphia—Nov. 9

Engineers' Club, 1317 Spruce St., Philadelphia, Pa.; dinner 6:30 p.m. Relationship Between Diesel Engines, Fuels and Lubricants—L. A. Blanc, Caterpillar Tractor Co. Technical chairman: Frank Nail, Mack Mfg. Co.

Salt Lake Group—Nov. 14

Hotel New House; dinner 7:00 p.m. New Developments in the Internal Combustion Engine—Russell G. Riley, director of merchandising, Thompson Products, Inc.

Twin City—Nov. 9

Solarium Room, Hotel Curtis, Minneapolis, Minn.; dinner 6:30 p.m. Coated Abrasives—James E. Trask.

Virginia—Nov. 21

Hotel William Byrd; dinner 7:00 p.m. Meeting 8:00 p.m. Safety on Motor Trucks—Charles Ray, director of safety engineering, Markel Service, Inc., Richmond, Va.

Williamsport Group—Nov. 7

Antlers Club, Williamsport, Pa.; dinner 6:45 p.m. The Thompson Vitameter—C. H. Van Hartesveldt, executive vice-president, Thompson-Toledo Vitameter Corp. Colored films and slides to accompany talk.

NATIONAL MEETINGS

MEETING	DATE	HOTEL
ANNUAL MEETING and Engineering Display	Jan. 9-13, 1950	Book-Cadillac, Detroit
PASSENGER CAR, BODY, and PRODUCTION	March 14-16	Book-Cadillac, Detroit
AERONAUTIC and Aircraft Engineering Display	April 17-19	Statler, New York
SUMMER	June 4-9	French Lick Springs, French Lick, Ind.

TRAILER MATERIAL Choice Can Net Payload Gains

BASED ON PAPER* BY

L. H. Chaillé and **V. H. Stewart**

Director of Public Relations

Resident Engineer,
Fruehauf Trailer Co. (Western Division)

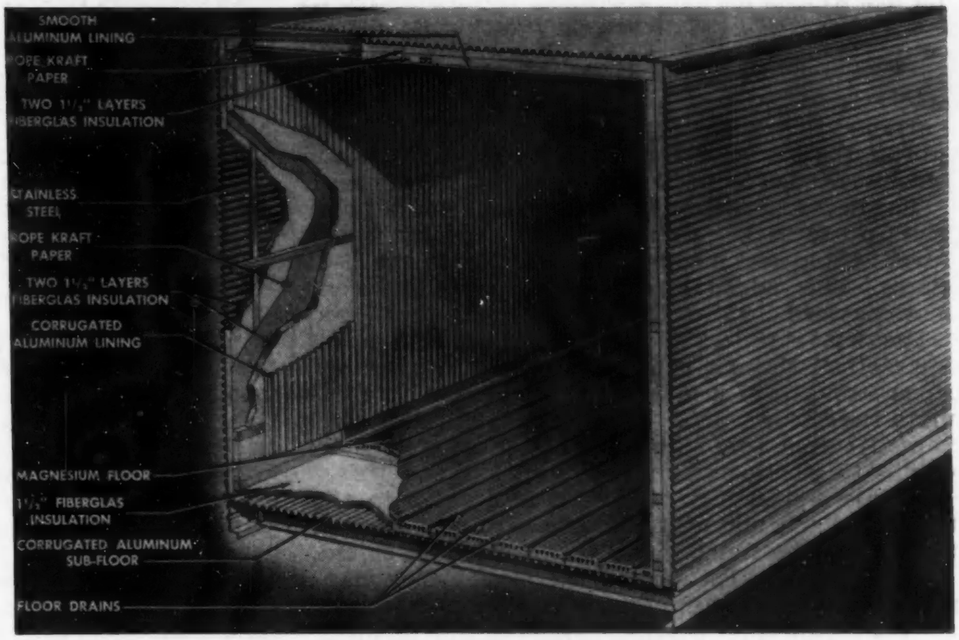


Fig. 1—This cutaway view reveals construction details of a refrigerator trailer in which stainless steel, aluminum, magnesium, and other materials are used for different purposes. These materials also are used advantageously in other trailer components

SKILLFULLY selecting the right metal for every part of a trailer can make a big difference in its efficiency and revenue-producing possibilities. Stainless, carbon, and alloy steels as well as magnesium and cast and forged aluminum are best for specific jobs in the trailer design, examination of a van-type trailer shows. See Fig. 1.

For example, basic material selected for the trailer van was an 18-8 stainless steel (18% chromium and 8% nickel). Used for side walls and roof, this material offers permanent rust and corrosion proof protection and the economy of freedom from painting. It permits beauty in structural lines and retains the enduring polish it is possible to give the material.

Stainless steel used for side and rear doors and nose of the trailer gives greater resistance to shock. The door panels are formed from sheets of stainless steel of a gage almost as heavy as the door-reinforcing members.

One main characteristic of stainless steel is that it becomes work-hardened much more rapidly than do ordinary steels. The more it is worked or formed, the stronger it becomes—up to a certain point. By cold working the steel into ribbed body panels and structural shapes, its tensile strength is raised to 150,000 psi and its yield point raised to 120,000 psi, more than twice that of ordinary steel. Fig. 2 illustrates the rib-forming process.

Since stainless steel is an exceptionally strong material, its qualities suggest many possibilities to the designer. Its modulus of elasticity remains reasonably constant for the complete range of alloy

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* Paper "How Would You Design a Trailer for Maximum Efficiency?" was presented at SAE National West Coast Meeting, Portland, Oreg., Aug. 15, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price 25¢ to members, 50¢ to nonmembers.)

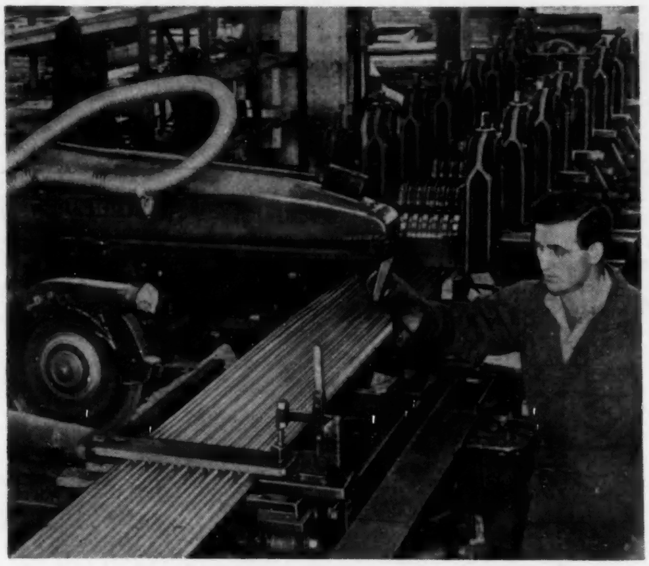


Fig. 2—By a progressive forming process, strong, thin sections of stainless steel are rolled into stiff ribbed sections for trailer sidewalls and roof. Such work-hardening imparts high strength to the material

steels, making possible weight savings closely proportional to permissable area reductions. In other words, the stronger the material, the smaller the amount needed to meet a specific stress requirement.

The new square-front stainless-steel van not only gives up to 29 more cubic feet of capacity, but it saves 2095 lb over a conventional trailer in the 35-ft class. This added load-carrying capacity means more payload for the operator.

Fact that stainless steel sections are so thin makes impractical the use of bolts or rivets in assembly, since strength of a material must always approximate the shear value. Fortunately, stainless steel is well suited to welding. This saves weight by eliminating bolts and rivets.

Stainless steel cannot do the entire job in the trailer. Each material must be carefully selected to perform the function for which it is best suited.

A case in point is the newly-developed alloy magnesium floor. Most important element of this floor are the magnesium boards. These are made up of a series of lengthwise "I" beams, the upper flanges of which are joined to form boards about 6 in. wide and as long as the trailer. The lower flanges of the "I" beams rest on the trailer cross-sills. One outer "I" beam has a tongue; the one on the other side has a groove. Tongues and grooves are made to interlock.

This floor also is adaptable to a refrigerator trailer by adding a keeper between the floor sections. The keeper provides a space for air circulation under the load, eliminating, in most cases, the use of wooden duct boards, which are heavy and difficult to maintain.

Specially-formed sidewall sections fit the outer boards and carry the magnesium floor several inches up the sidewalls, serving as built-in flashings. All areas that might be subject to corrosion are coated with zinc chromate paint.

Magnesium alloy used in trailer floors offers the advantages of high resiliency, for good capacity of energy absorption, and light weight. Weight saved in a typical 35-ft dry cargo van trailer is about 800 lb as compared to an ordinary floor; in a refrigerated unit as much as 1250 lb can be saved.

Aluminum also can be put to good use in certain trailer parts. Forged aluminum wheels can be provided as standard equipment on stainless steel trailers. They are stronger than cast aluminum wheels because the forging process gives the wheels a close-knit structure which is free from defects sometimes found in castings. These wheels are actually stronger and more durable than cast steel wheels, yet save 93½ lb per axle with 20-in. tires.

Cast, extruded, and formed aluminum is used in vertical supports which hold up the trailer nose when the tractor is uncoupled. Various types of aluminum are used in the components of the support assembly. The support legs are 61 ST aluminum, the support axles of 24 ST aluminum tubing, and the wheels of 3/16 in. drawn aluminum, heat-treated to 61 S-T6 after drawing and welding. Aluminum supports save 120 lb in weight over conventional assemblies.

In extensive tests conducted to compare aluminum and steel trailer landing wheels under compressive and impact loadings (subjecting each wheel to loads up to 47,200 lb), performance ability of the aluminum wheel compared very favorably with that of steel.

As to advantageously using steels other than stainless, one application is for kingpins, used to connect trailer to tractor coupler. Alloy steel—heat-treated, precision machined, cold riveted, and welded to the coupler plate—produced a kingpin strong enough to pick up a locomotive.

The upper coupler apron plate is formed from hard carbon steel. This is especially designed to provide a wide surface so that tractors may back into the trailer from an indirect angle. Carbon steel has good wearing qualities. The continual friction on this plate plus a certain amount of grease makes a rust-resistant material unnecessary.

High-tensile steel is used for coupler backing plate members. Their box section construction permits them to take the buffeting of the coupling.

Best bet for an axle especially strong vertically, where loads are heaviest, and with useless weight eliminated horizontally, would be an "I" beam axle. Made of double heat-treated chrome molybdenum steel, this axle is rated at 20,000 lb capacity, providing a 2000-lb greater margin of safety over axles of other types rated at 18,000 lb.

Because of its unique design, this axle is stressed to only 17,000 psi, yet the material has a yield point of 110,000 psi. Result is a safety factor of almost 7 to 1. This axle actually weighs 75 lb less than a tubular axle in popular use.

An ideal material for braking surfaces in brake drums is nickel alloy cast iron. Pressed steel brake shoes save an additional 80 lb per axle. Maintenance records of many fleets reveal that this combination offers an efficient brake with the advantage of low maintenance cost.

There are many other materials that go into making an efficient trailer. Corrugated sheet aluminum or plywood is used for lining. Rubber bumpers are placed on the rear for dock protection. And cast aluminum hinges are used for lightness with strength.

HIGHER PRODUCTIVITY

Sought at Record Tractor Meeting

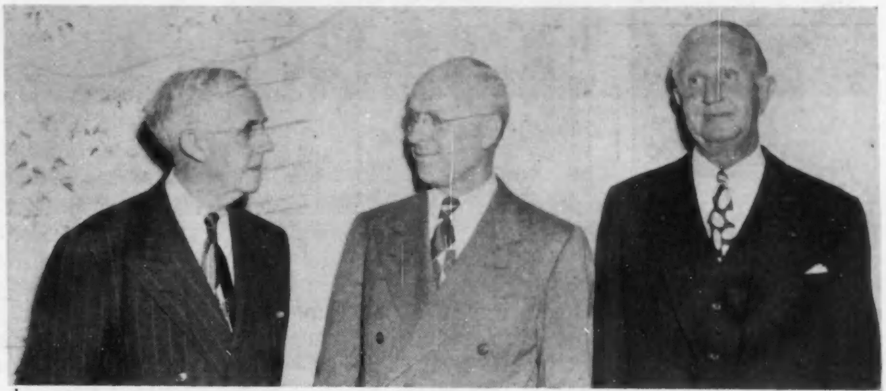
"AS ye sow, so shall ye reap" prophesies rich rewards for engineers at the SAE National Tractor Meeting at Milwaukee, Sept. 13-15, who laid a groundwork of fertile ideas aimed at cultivating longer-lasting and more productive tractors, implements, and earthmoving machines. Promise of better things to some in farming and construction equipment stemmed from an interchange of new knowledge on what these machines do to the earth, and what the earth does to the machines.

An all-time high in attendance of 739 made this the biggest SAE National Tractor Meeting, with high technical session attendance indicative of interest in the program.

At the dinner, 317 members and guests heard John L. Collyer, president of the B. F. Goodrich Corp., urge continued use and improvement of man-made rubber. He also praised the SAE Tractor Technical Committee for its standardization and simplification of tire sizes, and told of the progress in tires for earthmoving machines. Standardization work of the SAE Construction and Industrial Machinery Technical Committee, was served up by E. F. Norelius as an added event.

Demands of the customer—the excavation contractor in the earthmoving machine field and the farmer in the tractor-implement area—to bring down costs per unit of work performed by their respective machines are being better fulfilled, engineers at the meeting said.

Earthmoving machine designers, shooting for lower costs per cubic



DINNER SESSION participants included (left to right) SAE President S. W. Sparrow, SAE Past-President C. E. Frudden, toastmaster, and dinner speaker John L. Collyer

Dinner Speakers

SPARROW said in part:

"Without claiming perfect vision, I can see definite indications of continued growth in size, in strength, and in the value of SAE to each member. I like the word 'growth' better than 'expansion.' Expansion brings to mind a toy balloon and the discouraged piece of rubber that remains when the expansion has been too great. We don't want that type of expansion.

"We don't want a flock of new activities created by a puff of enthusiasm, but without permanent value. Nor do we want standards or other products of our technical committees that have not received careful and competent consideration. There is not much fear on that score.

"The notion that there are two sides to every question did not originate in SAE. If there are 10 members on a committee, then there are 10 sides to every question and much to be said for each side. What is more, it usually is said—and repeated.

"If the 'common sense of most' continues to control (in SAE), then the present is secure and the future is bright."

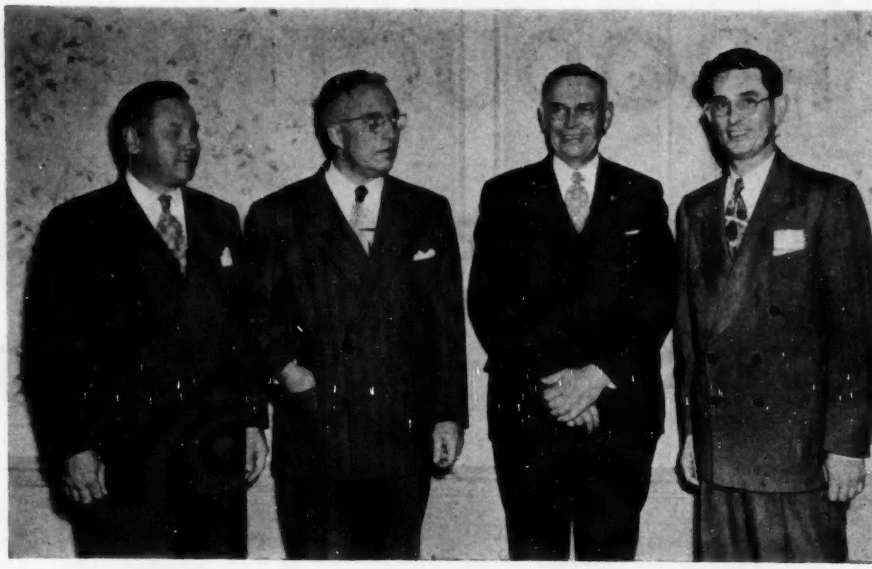
COLLYER said in part:

"A yardstick of the importance of rubber to our economy is the spectacular increase over the years in the consumption of rubber. . . . In 1870, the consumption of rubber in the United States totaled only 4000 tons. The peak prewar consumption was in 1940 with a total of 650,000 tons.

"The war period is a story in itself; but moving to postwar, we find that for 1946, 1947, and 1948 consumption of rubber was in excess of a million tons. Moreover, we estimate that consumption of rubber this year will be close to a million tons, providing no prolonged stoppages of work occur in our industries.

"So you see that postwar consumption is running at a rate 250 times the level of 1870, and 50% above the highest prewar rate. Economic progress of the United States has been the most important factor in this enormous growth.

"This year consumption in the United States will be almost 14½ lb per person, while per capita consumption for the rest of the world amounts to only about 0.9 lb."



Among those who helped make the SAE National Tractor Meeting a success are (left to right) C. A. Hubert, chairman, SAE Tractor & Farm Machinery Activity Meetings Committee; L. A. Gilmer, SAE Vice-President for Tractor and Farm Machinery Activity; G. W. Curtis, chairman of the General Committee for the meeting; and G. J. Haislmaier, chairman of SAE Milwaukee Section

yard of dirt moved, checked on merits of increasing horsepower, capacity, or top speed and came up with cost figures revealing the best compromise for various operating conditions.

For comparative costs, engineers at the meeting started with a basic rubber-tired tractor and scraper unit and studied it with three design changes—60% increase in horsepower, 60% increase in capacity, and 60% increase in top speed. Up to 1000-ft hauls on 1¼% downgrades, they found the increased capacity design is the greatest producer, although it has the lowest average speed and the highest fixed cycle. For hauls longer than 1000 ft, the unit with greater horsepower shows an increasing production advantage.

In other assumed operating conditions as well as this one, the high-horsepower and high-capacity units show up advantageously from a production as well as cost standpoint.

Job Flexibility a Criterion

Versatility of earthmoving machines was labelled as another significant feature. Giving a contractor equipment that will fit the greatest variety of jobs and conditions to meet competition, engineers agreed, is one of their biggest responsibilities.

Smaller and medium size units are more versatile for mass earthmoving production, some maintained, because more units may be used on long hauls to keep the pusher or shovel busy. Haul road maintenance for getting higher average speeds is more economical with smaller and medium size units, not to mention transportation from job to job and investment in standby equipment.

One of the most versatile machines was said to be the crawler tractor-scraper unit. This is a universal tool, considering the various types of dirt-moving operations.

It was pointed out that the average dirt haul is less than 2000 ft long. A project may have some hauls longer than this, which forces the contractor to decide whether to purchase or rent the proper equipment or to accept a higher cost and use standard equipment. Fact that the choice is voluntary and not forced on the contractor equipped with crawler machinery is a point in his favor.

Rubber-tired equipment proves satisfactory for normal ground conditions, admitted crawler advocates, but is immobile when the ground is slick after a rain. The crawler tractor can travel under extremely bad conditions. High power of the rubber-tired unit compared with struck capacity represents high fuel costs per hour.

However, some engineers argued in favor of greater horsepower in earthmoving machines for higher speeds, within limits of road conditions and personnel safety.

Typifying this approach is an 18-yd struck bottom dump tractor described, powered by two 190-hp engines, each equipped with a torque converter and transmission. This amount of power, applied to four driving wheels, coupled with ability to shift the transmissions simultaneously under full torque, evolves a unit which performs well in soft, sandy conditions. This machine not only ascended a 15 to 20% grade built of wet sand, but also pushed a 13-yd unit ahead of it.

Engineers argued that two engines on the same unit are justified because lower cost per horsepower results (higher production engines are used), the powerplant can be placed relatively close to the point of application, better weight distribution is possible, and clutches, transmissions, and propeller shafts cost less.

Hydraulic Controls on Increase

The higher productivity aim is fast making hydraulic power an integral part of earthmoving machines, examples cited at the meeting indicate.

Finer control of bulldozer blades and power shovels with less operator fatigue were among the advantages claimed for hydraulic power. The hydraulic equipment trend also was said to be carrying over to scrapers and grades. For years a single-purpose machine, the road grader today can be adapted to snow plows and loader attachments using

air hydraulic circuit with simple piping and low conversion expense.

Farm equipment engineers reciting the performance of their designs on tillage, cultivation, and harvesting laid down demands being made on their machines by modern farming techniques. New cultivating methods were said to require an ever-increasing range and flexibility in tractors which will satisfy the farmer, "who values his tractor for the work it will do, the loads it will push, pull, lift, and carry in good and bad footing, for the machines it will drive, and the places it will go without damaging crops."

Lower speeds are going to be necessary, implement designers predicted, to handle high-yielding hybrid grain and cotton varieties—and to power corn pickers built to handle "picket-fence" rows of corn, planted in widely-spaced rows . . . but they warned, farmers also want to work faster and handle more acres per hour.

Desirable, too, will be increased draft and power take-off capacities on small as well as large tractors, because the small farmer wants to contour his farm with his light equipment—and insists that he be able to maintain terraced fields and do some bulldozing and grading jobs, however slowly.

Higher clearances—and perhaps cab-over-engine designs—also were suggested by implement men as needed to accommodate the equipment used in spreading anhydrous ammonia fertilizer, and to better balance the heavy equipment carried in the rear for modern spraying, flame throwing, soil fumigation, and liquid fertilization.

Reemphasize Basic Jobs

But one tractor engineer reminded his colleagues that the farm tractor's greatest use is in tillage, cultivating, and harvesting operations, and that efforts should be made to adapt the design to as many requirements as possible, but not at the expense of features necessary for these basic jobs.

All this and the "heaven too" of low price and economical operation seems to be the requirement, according to implement designers who spoke at the meeting. "All we implement designers want," one of them summarized, "is for the tractor engineer to shrink his powerplant and gear trains into some out-of-the-way corner, and pull more power out of the assembly at convenient points!"

Conservation farming (preserving the soil and making it resist erosion) was said to be changing farming operations—which in turn also change the traditional requirements for farm machinery. Reiterating these modern maxims, soil conservation experts revealed that experience and research has gone a long way to spell out the "how" of these changes.

The brunt of these changes, comments indicated, fall on the farm implements, but discussion showed clearly that tractor design, too, will be challenged, even though indirectly through demands made by implement changes.

Need for permitting fields to be bare for as short a time as possible, for example, brings need for implements which have greater maneuverability and ease of operation, tractors on which implements can be mounted and changed more readily, smaller sized units, more adaptable units, and greater availability of dirt-moving attachments.

Needs for producing a seed bed which avoids highly pulverized surfaces brings other new implement requirements, as does the growth of mulch farming and other soil conservation practices. And in each case, the specifics of implement design needed are pretty well known. Many of them were detailed by experts for the tractor-implement audience at the meeting.

The change from rectangular-shaped fields to long, narrow contour strips, called "the transition from the compass to the bubble tube in determining row direction," also was seen as a challenge to implement and tractor design. Corn pickers able to negotiate sharp radii of curvature, improved machinery for terrace construction, and hydraulically-steered tractors for precise contour row cultivation were cited as growing needs in this area of conservation farming.

Machine Research

Second phase of their problem, meeting participants noted, is the effect of the earth (and its products) on the machines that work it. Working forces and induced stresses are giving way to studies relying on instrumentation and stress analysis, which effort, researchers predicted, should pay off in longer-living machines.

For example, tiny, sensitive instruments are revealing the behavior of earthmoving behemoths, engineers at the meeting revealed. Performance of

Under the general chairmanship of C. W. Curtis, the following served as chairmen of the six technical sessions of the SAE National Tractor Meetings: **J. W. Bridwell, R. A. Beckwith, H. L. Brock, W. H. Worthington, D. C. Heitshu,** and **W. E. Knapp.**

This report is based on discussions and 13 papers. . . . "Obtaining Higher Average Speeds by the Utilization of Horsepower in Earthmoving Equipment," by **John P. Carroll,** Caterpillar Tractor Co.; "Utilization of Horsepower in Earthmoving Equipment—Use of Large-Capacity, Slow-Speed Equipment," by **John E. Marson,** Bucyrus-Erie Co.; "Utilization of Horsepower in Earthmoving Equipment—Use of Higher Maximum Speeds," by **Leslie Rittenhouse,** The Euclid Road Machinery Co.; "Hydraulic Power as Applied to Earthmoving Equipment," by **E. J. Hrdlicka,** Hydraulic Equipment Co.; "Research on the Performance of Power Cranes and Shovels," by **Trevor Davidson** and **J. H. Meier,** Bucyrus-Erie Co.; "Dust and Its Effect on Air Cleaner Design," by **W. W. Lowther,** Donaldson Co., Inc.; "Lubricant Retention and Dust Exclusion on Farm Implements," by **S. C. McCombs** and **R. O. Isenbarger,** Chicago Rawhide Mfg. Co.; "High Frequency Heat-Treatment of Gears—Equipment and Processes," by **J. A. Redmond,** Westinghouse Electric Corp.; "Induction Hardened Gears," by **H. B. Knowlton** and **H. F. Kincaid,** International Harvester Co.; "Tillage Forces and Their Effect on the Farm Tractor," by **A. W. Clyde,** Pennsylvania State College; "Gear Drives in Implement Design," by **E. E. Eaton,** Clark Equipment Co.; "New Cultivating Methods," by **S. D. Pool,** International Harvester Co.; and "Soil Conservation Practices as Related to Farm Tractor and Implement Design," by **G. E. Ryerson,** Soil Conservation Service, U. S. Department of Agriculture.

power shovels and draglines in removing clumps of clay and outgrowths of rock as well as the reaction of their parts to high repetitive loads in these operations are emerging from such test studies. One of the devices used is a bonded wire electric strain gage. Ideally suited to such dynamic work, the strain gage transmits data to recording parts remote from moving parts.

Stress determinations made in this way, specialists said, establish the suitability of a part for the service to which it is subjected. Also it is possible to determine the stress imposed upon a part by service loads.

Another delicate instrument being put to work to study the behavior of a shovel as it takes a cut out of rock, clay, sand, and other materials is an electric pantograph. It plots power data against dipper position and also the path of the dipper. By laying a scale model of the dipper and handle on the trace, the angle of attack between dipper and bank can be studied.

Tractor and implement engineers also were urged to learn more about the forces encountered by a tool working the earth. They heard that more information on the behavior of these tillage forces is a must to supplant cut-and-try methods of designs.

Implement designers admitted the need for knowing more about these forces for developing the tool, in operating or hitching to it, and setting up shop tests that will duplicate service loads on the tool, or part of it. Tractor engineers declared such information valuable for determining stability and traction.

Design theory related to tillage forces was called inadequate on two counts. First, engineers must get straight on the use of mechanics in dealing with the forces involved, a theoretician said. Second, information is yet lacking on the vertical and side forces on tool and their relationship to the draft.

Tractor men told how they were meeting these high working stresses in gears with high-frequency induction hardening to increase the strength and lower the cost of these components. Hailed as a boon to heat-treatment, this one-at-a-time method of hardening was said to cost less for labor and equipment, compared with carburizing, if the production is reasonably high. The one-at-a-time system also saves handling and trucking costs. The gears can be hardened as they come off gear-cutting machines one at a time, or in small lots. This technique also was claimed to eliminate a large backlog of gears and costly furnace shutdowns.

Permit Less Costly Steels

But principal economy was said to stem from the replacement of alloy steels with carbon steels. Given as an example of this feature of induction hardening was a large 36-in. diameter, 6-in. wide final drive gear. Formerly made of nickel-alloy steel, induction hardening permitted a change to carbon steel, with 15-lb saving of nickel per gear. The higher hardness imparted to the gear made for greater load-carrying ability and wear resistance. In fact, the gear width could have been reduced and the performance of the nickel-alloy gear maintained.

Other advantages claimed for induction heat-treatment include the elimination of machining operations (needed to correct distortion which accompanies other hardening methods), and the cleaner, cooler working conditions with this process.

Turning from the pro to the con side of induction heating, engineers pointed out that induction heating equipment requires specialized maintenance and rigid metallurgical control. Many applications were said to involve substitution of higher carbon steels, requiring special attention to machining methods, and in some cases shortening tool life and lowering production. Final score showed most in favor of induction heating, feeling that its advantages far outweigh the problems it presents.

Gear production discussions veered off to the design side, with implement engineers forecasting simplification and complete enclosure of gear drives.

They decried current drives that are open, exposed, and over-elaborate. These designs require too much of the farmer's time for lubricating and servicing them. Greasing instructions for one such drive calls for lubrication of 62 individual points every 4 hr of operation.

Another current problem in this area, brought to light at the meeting, is the wide variety of gear drive designs being produced for accomplishing identical functions. Even for the same type equipment, different gear drives can be found. While this is complicated by the number of motions and speeds these drives must impart, it does create a problem in the field.

Standardization Urged

Engineers at the meeting saw the demand for greater interchangeability bringing on standardized units. Desire for lower unit cost and improved service facilities, they continued, will serve as a further impetus toward standardization.

Dirt and earth in small quantities can do more harm to mechanized equipment on its finely-machined internal surfaces than can large soil masses resisting working tools, air cleaner and oil seal, engineers warned.

Tractor men were told that protection of engines in farming operations depends on choice of air cleaner and its installation. Most important air cleaner feature, they learned, is adequate capacity. Inadequate capacity causes dirty oil to be pulled over into the engine from the air cleaner. Dust-laden oil condenses in the air cleaner element. Here the dust settles out and the oil is used over and over again in the wetting process of trapping dry dust particles.

Since the dust-laden oil must return by gravity, there is a very definite speed through any air cleaner at which the oil will be pulled over. For this reason, the air must flow safely below this speed under all engine air demands.

Dust capacity of the air cleaner also emerged as a significant factor in cleaner selection, since the operator does not want to be bothered with servicing, except at specified times. The oil cup must have an ample oil supply to keep plenty of fluid oil after its quota of dust has been reached. The overall restriction under these conditions should be close to the initial reading.

Implement and tractor designers argued the merits of synthetic rubbers versus leather for seals

to exclude dirt, water, and other foreign matter from bearings and bearing surfaces.

Cited as an advantage of synthetic rubber seals is the fact that they will withstand higher temperatures than leather—275 F as compared with 225 F. It was also said that synthetic rubber seals, being dense and nonporous, will satisfactorily seal low-viscosity fluids, which tend to penetrate and seep through the fibrous structure of leather.

Synthetic rubber seals also can be molded to shapes and designs, permitting maximum reduction of running friction and the resultant temperature rise.

Among the disadvantages of synthetic rubber is the tendency of some compounds to adhere to shafts or hubs during long periods of machine idleness. It also has been found that to function efficiently and to have long life, synthetic rubber seals must operate on much finer shaft finishes than leather. And prelubrication of synthetic seals at time of assembly is vital to insure against scuffing and premature wear.

Seal designers cautioned that the advantage of synthetic rubber as regards sealing low-viscosity fluid can boomerang, since synthetic seals must have adequate lubricant supply at the sealing lip at all times during operation . . . the nonporous character of synthetic permits no absorption and redistribution of lubricant, which takes place with leather seals.

Annual Meeting

Book-Cadillac Hotel • Detroit

of the Society of Automotive Engineers

Jan. 9-13, 1950

Aero Meeting Studies

THAT producibility and performance of aircraft must and can advance hand in hand was an inescapable conclusion of the National Aeronautic Meeting.

SAE's best attended postwar aeronautic meeting drew 800 engineers to Los Angeles for four lively panel discussions on producibility, one on the transport of 1955, and six technical papers. Many arrived early to spend Wednesday, Oct. 5, inspecting the Southern California Cooperative Wind Tunnel and Hydrodynamic Laboratory at California Institute of Technology. And more than 450 remained for the Saturday evening dinner, making it a sell-out.

All 800 at the meeting went away conscious that the tempo of modern warfare now forces the military services to define "producibility" and rate it on a par with "tactical suitability" and "engineering" in the evaluation of new aircraft designs.

Reflections of the military policy came in three panels that considered means which designers, production experts, and management can use to achieve ready producibility plus high performance in aircraft and engines. The panels generated an awareness that provision for producibility must start as the prototype takes shape on the drawing board, and continue through production planning right up to production testing of the peak volume.

The aim of producibility—availability of high-performance aircraft in time of national emergency—calls for standardization of military and commercial transport designs, a fourth panel agreed. If a fifth panel which appraised the 1955 transport is right, operators will be standardizing on a turbojet or turboprop airliner in the 450-500 mph class, costing around $2,000,000 per airplane if produced in 100-unit lots.

Assurance of some of the improvement in performance that the industry seeks along with producibility came in reports of a source of more-reliable cascade test data, new knowledge on turbine engines' tolerance of sand and dust, research on fuel spray nozzles, an improved magnesium extrusion alloy, an all-pneumatic auxiliary power system, and a wheel-slide protection device.

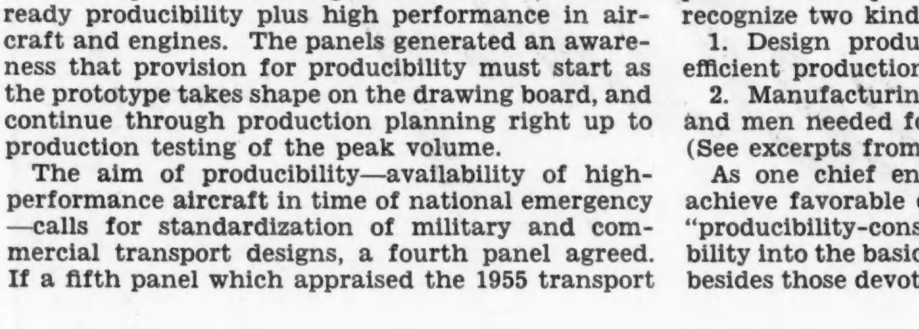
Producibility Plus Performance

Engineers at the meeting learned that the military services want all the producibility they can get without sacrificing performance gains. To evaluate producibility of proposed aircraft designs, the Air Force plans a new producibility specification. It will recognize two kinds:

1. Design producibility—Does the design allow efficient production?

2. Manufacturing producibility—Are the facilities and men needed for efficient production available? (See excerpts from discussion on pages 17 and 18.)

As one chief engineer explained it, the way to achieve favorable design producibility is to employ "producibility-conscious" engineers to get producibility into the basic design; then to assign designers, besides those devoted to the experimental schedules,



Ready to explore subject of optimum engine producibility are panel members W. P. Cross, Pratt & Whitney Aircraft Division, UAC; W. C. Heath, Solar Aircraft Co.; R. P. Kroon, Westinghouse Electric Corp.; A. T. Colwell, Thompson Products, Inc.; K. N. Bush, General Electric Co.; and Panel Chairman R. F. Gagg

PRODUCIBILITY

". . . quality enabling a design to be produced with minimum expenditure of materials, time, and labor. . ."

to work on producibility of design details and to explore manufacturing possibilities.

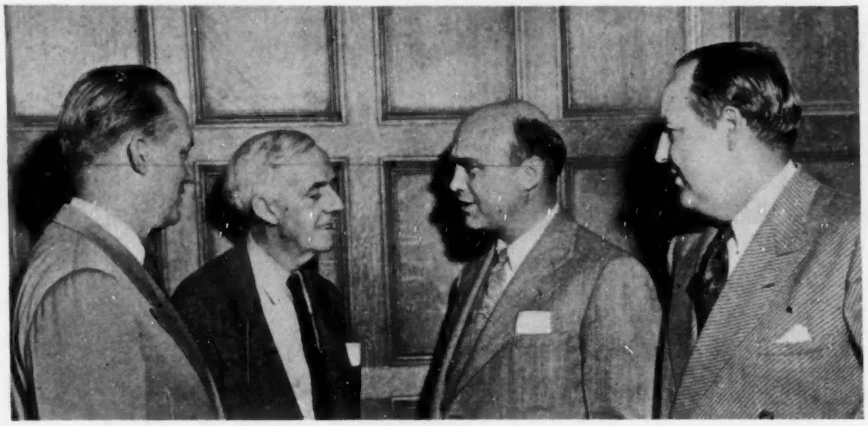
There was a lesson in producibility for these designers in a breakdown description of the J34. This turbojet is designed to be built up from seven major subassemblies. The design avoids close tolerances between mating parts. For example, the slender, flexible turbine shaft permits considerable misalignment of the stationary parts without unduly large bearing forces or shaft stresses. Such design details as quick-disconnect joints and harnesses holding all electrical conduits within a common covering facilitate assembly.

Because of the J34's subassemblies, assembly teams can specialize. Their training takes less time, which increases producibility. And their greater skill helps performance.

Assigning extra designers to get this kind of producibility into the design during prototype stages adds to costs. Future prototype contracts must

Production and design experts (shown left to right): Major-Gen. K. B. Wolfe, USAF, Deputy Chief of Staff for Materiel; S. J. Pipitone, Glenn L. Martin Co.; G. W. Motherwell, Wyman-Gordon Co., prepare to explain significant structural design and fabrication developments. Panel Chairman F. L. Magee stands at the lecturn

Panel discussion on basic problems of producibility brings together (left to right) Harold Harrison, who presented discussion by H. L. Hibbard, Lockheed Aircraft Corp.; Rear-Adm. A. M. Pride, USN, Chief, Bureau of Aeronautics; Brig.-Gen. A. H. Johnson, who presented discussion by Major-Gen. F. M. Hopkins, Jr., USAF, Chief, Industrial Planning Division, AMC; E. B. Newill, Allison Division, GMC; and Panel Chairman F. C. Crawford



D. Roy Shoults (second from right), general chairman of the producibility panels, accepts congratulations of SAE President Sparrow on the four well coached discussions, while C. F. Thomas (extreme left), general chairman of the meeting, and Peagan C. Stunkel (extreme right), chairman of the host section, Southern California, look on. Also cooperating were SAE's seven other West Coast sections and groups and the Aircraft Industries Association of America, Inc., and the Air Transport Association of America

cover the added costs, it was pointed out.

Even when designers must scarifice producibility in interest of performance in the prototype, continued effort can improve both in the light of prototype testing, an engine builder showed. When an engine structure is first conceived, the designer usually employs the best of materials to assure highest strength in spite of elevated operating temperatures. (He may use materials like columbium, cobalt, and chromium, that are too scarce for volume production of engines.) But testing experience shows how to eliminate the vibrations that underlie the strength requirements.

Then the structure can be redesigned for fabrication out of ordinary materials—making it more readily producible. And the structural improvements better the engine performancewise.

New Production Techniques

Panel members disclosed new techniques and tools that improve manufacturing producibility of turbine engine blades and sheet metal parts. Simultaneous improvements in accuracy insure that the engines will attain designed-in performance.

For example, frozen mercury patterns make possible precision casting of turbine and compressor blades too complicated to make any other way, it was reported. The process depends on the fact that two clean surfaces of solid mercury will weld together with very light pressures. Building up complex mercury patterns is relatively easy. Completed patterns are dipped in ceramic, forming molds much thinner than those formed by investing wax or plastic patterns.

Cores for hollow castings can be built up with the mold during the ceramic coating operation. This way, cores can be formed thin enough to offer little resistance to the cooling metal around them. Distortion of hollow blades is minimized.

Hollow, internally cooled blades made by this process might permit considerably greater power output because they could tolerate higher gas temperatures, it was pointed out. Or, operated at present peak temperatures, they could save on critical high-temperature materials.

Surface finish of parts made by this process is excellent. Dimensional tolerances can be held 50% closer than with the wax method. Although experts dispute the magnitude of importance of these two factors in blade performance, they agree that finer finishes tend to improve performance and do reduce stresses.

Simpler, solid blades are most readily producible by precision forging, rolling, and the powered metal process, according to a blade manufacturer.

Producibility of sheet metal parts, like exhaust cones and burner liners, is profiting from direct-current, dry-disc resistance welding machines featuring a low inertia head for fast follow-up pressure. The head cuts scrap losses, saves time, and improves parts by minimizing internal cracking, gas porosity, shrink voids, and expulsion of metal, according to its proponent.

Important to any method of fabricating sheet metal engine parts are machine feeding and indexing fixtures, he told production specialists.

The process of achieving producibility extends even to production testing of the finished article, a test expert showed. Testing involves test cells, fuel, and manpower—all of which would be critical in time of national emergency. His examination of piston-engine testing practices disclosed the secret of efficient testing: thorough training of test personnel, attention to preparing the engine for testing, and preventive measures to avoid need for engine repairs during the course of testing

J47's from Subcontracted Parts

Along with these specific tips for production engineers on increasing manufacturing producibility came a lesson for management in the story of the J47 turbojets produced at Lockland, Ohio, entirely of parts supplied by over 200 subcontractors. The prime contractor merely specifies the parts to be purchased, checks delivered parts, and assembles them.

This policy of subcontracting perpetuates production skills in plants built up during the last war and decentralizes strategic industries. In case of emergency, it will be easier for each subcontractor to expand and new ones to join than for one complete plant to expand.

In the manufacture of airframes, producibility is catching up with performance of new wing designs,

a team of design and production specialists demonstrated. They explained that to take advantage of the extra performance that jet engines can give airplanes, wings had to be thinner and more highly loaded. Keeping weight of the thin wings to a minimum required continuous thick tapered cover material. Now with new facilities for forging aluminum and magnesium, the new thin wings will be more readily producible in large quantities than old-type wings were.

The thin wings resulted from a new design philosophy which substitutes elastic stability theory dealing with plates for that dealing with columns for determination of the allowable critical compression load on the upper wing cover. Thus critical load is an inverse function of the square of the spacing between beam webs and is independent of rib spacing. Therefore, as far as compression load is concerned, the ribs can be eliminated.

Eliminating the ribs enhances producibility and performance. There are fewer parts to make and fewer to assemble. The finished wing weighs 10% less and is more accurate because there is no accumulation of tolerances of internal structure.

To get along without ribs, the wing cover must have bending and shear stiffness in the chord direction as well as in the span direction. (Former covers required spanwise stiffness only.) That means covers must be continuous in the chord direction as well as in the span direction. But the amount of material required decreases toward the tip; the weight efficiency of the new covers depends on tapered blankets.

Tapered cover can be machined from rolled or forged flat stock, or it can be rolled or forged tapered. For large scale production and flexibility in detail design, forging is the best process, according to one expert. He recommended that suppliers of forged sheet direct their development toward longer, wider, thicker sheet material and leave the tapering to machinists.

Forging aluminum and magnesium aircraft components requires the slow squeezing action of hydraulic presses. (These metals do not respond satisfactorily to the impact forging equipment usually used for ferrous metals, it was explained.) Because the light metals can be forged at relatively low temperatures, around 800 F, die temperature can be maintained close to forging temperature, thereby reducing heat loss during the slow press action.

Pilot Forge Plant Begun

The Germans pioneered use of large pressure forgings for magnesium. They built three 15,000-ton and one 30,000-ton presses and were well along on design of a 55,000-ton press when the war ended, it was reported. War demonstrated the advantages of large presses to the USAAF. In 1944, the United States government financed construction of an 18,000-ton press near Worcester, Mass. Now, in addition, American industry is acquiring German equipment. Some will go to a government plant in Adrian, Mich., set up to study forging techniques.

Maintaining this design- and manufacturing- producibility potential disclosed at the meeting will require sustained procurement, a manufacturer's representative stressed. Speaking of engines, he said that if we are to have suitable designs when we need them, we must design, develop, manufacture and prove in flight continually advancing types. Readily producible engines can be designed, even if quantities on contracts are small. It is continuity of work—not necessarily large orders—that insures performance with producibility, he said.

One step toward providing this continuity was

Panel waits for audience's reaction to this year's appraisal of the 1955 air transport. Panel members are (left to right) D. J. Jordan, Pratt & Whitney Aircraft Division, UAC; Carlos Wood, Douglas Aircraft Co.; H. R. Harris, American Overseas Airlines; E. H. Atkin, A. V. Roe Canada, Ltd.; and Panel Chairman W. E. Littlewood

Members of the panel on interchangeability of military and commercial transports are pleasantly surprised to find themselves so much in agreement. Shown are (left to right) Major-Gen. L. S. Kuter, Commanding General, Military Air Transport Service; W. W. Davies, United Air Lines, Inc.; G. W. Haldeman, CAA; W. E. Beall, Boeing Airplane Co.; and Panel Chairman D. W. Rentzel

Left to right: success of the meeting brings smiles of satisfaction to Karl Arnstein, SAE Vice-President representing Aircraft Activity, R. C. Loomis, SAE Vice-President representing Air Transport Activity, E. G. Haven, Aircraft Powerplant Activity Meetings Chairman; and L. R. Koepnick, Air Transport Activity Meetings Chairman

Manly Memorial Medalist Andrew Kalitinsky (left) poses with Gaylord Newton, who made the presentation to Kalitinsky for his paper "Atomic Power and Aircraft Propulsion"

taken at the meeting. Military and commercial operators and manufacturers of transport aircraft agreed that standardization of military and commercial transport aircraft is practical and essential.

Although the military underwrite development of the costly new designs they must have to protect the nation, they cannot afford to procure a large standby fleet for quick mobilization. Commercial operators can't pay for continuing development, even though they need the economy it brings. But they have to buy the fleets anyway. Both sides reason that only by combining can the nation have a fleet of the most advanced aircraft on hand for emergencies plus its peacetime advantages.

By combining, operators expect also to increase expansibility of production rates, reduce production costs, and establish pools of trained workers all the way from the design board to the cockpit controls.

Although operators recognize differences in military and commercial certification codes, they hope to reconcile them. Some believe that the C.A.R. might serve as the basis for standardization, with the military using heavier loadings or shorter fields at the sacrifice of a little of the performance written in for the airlines.

Reappraisal of the requirements for the air transport of 1955 resulted in little change from the conclusions reached in a similar panel discussion at the October 1948 SAE National Aeronautic Meeting.

Turbojets vs. Turboprops

Turbojet enthusiasts—and they were again in the majority—talked confidently of vibration-free jet transports whose engines would consume only 0.9 lb thrust per hour. One engine-performance authority granted that the resources pouring into jet development will produce good jet engines for 1955 fighters and bombers, but he hinted that they might not be the optimum engines for transports. Turboprops are worth considering, he suggested.

An analysis of traffic potentials, stage lengths, speeds, costs, and airports indicated that the 1955 transport must be usable for trip lengths of 830 to 3500 miles in order that a production quantity of at least 100 airplanes may be set up. At that rate, the transport will cost about $2,000,000. This is the combination of maximum allowable price and minimum quantity necessary to put operating costs into a profitable region.

Operators are still seeking better visibility, less vibration, and more safety—all in an airplane permitting lower fares. They would like to raise last year's speed specification of 400 mph to 450-500 mph.

Promise of performance advances emerged from disclosures of new design knowledge and new products.

Two researchers described a new twist in cascade testing that yields more-reliable data for the systematic design of axial-flow compressors and turbines. Past data contained discrepancies arising

from interaction and interference of the boundary layers on the side walls of the tunnel with flow about the test airfoils. Now NACA solves the problem by drawing off the boundary layers through porous side walls installed in their tunnels, they explained.

Data presented on Air Force experiments with turbine engines operated under simulated dust-storm conditions pointed to gradual performance deterioration rather than rapid failures. Designers were advised to locate duct entrances in areas of low dust concentration or to use flash-type entrances. Sand removal from intake air appears hopelessly detrimental to engine efficiency, they were warned.

Since small amounts of sand ingestion left the engine much cleaner than at the start of testing, this was suggested as a cleaning method. Discussion brought out that the British have a liquid cleaner, which can do the cleaning without the erosion danger of sand cleaning.

Consideration of fuel spray nozzles for turbine engines revealed that it is the low-delivery range, corresponding to starting and high-altitude idle operation, that is troubling designers now. In this range, engine performance is particularly sensitive to degree of atomization and evenness of fuel division, they found. Duplex or expanding swirl-type nozzles give good atomization, good air mixing, freedom from seizure, and adequate low-delivery-rate performance, they concluded.

New Products Described

Recommended to designers of extruded airframe parts was a recently developed magnesium alloy containing 0.6% zirconium and 5.7% zinc. The zirconium refines the grain, improving strength, toughness, and notch-insensitivity. Beams of this alloy, designated ZK60, are efficient and readily producible. They can be made 25% cheaper and 5% lighter than comparable aluminum beams, said ZK60's advocate. He asserted that corrosion resistance of ZK60 compares favorably to that of some common aluminum alloys.

A user confirmed the advantages claimed for the alloy and reported that recently conducted gunfire tests indicate that ZK60 extrusions can withstand combat conditions satisfactorily.

Two new aircraft accessories were described, one to conserve propulsive power and the other to brake it more safely during landings.

An "all-pneumatic" system combining gas turbine compressor units and supercharger air extraction from the main engines was proposed as a solution to the problem of powering aircraft accessories and starting the main engines. Load demands at low altitude are met by the auxiliary unit and at high altitude by the main engines or main engines plus auxiliary unit.

The idea is to duct the output from the turbine-powered auxiliary compressor and from the main engines to a common manifold. Manifold air goes to a burner, then to power turbines which convert power from the compressed air to drive alternators and generators. These supply electrical power needs.

Exhaust heat of the turbine unit heats the cabin. Compressed air furnished to an air-cycle refrigeration unit cools it.

For wing deicing compressed air bled through the nozzle of a jet pump induces airflow through a combustion heater. For starting the main engines, the auxiliary unit compresses air which passes through the manifold to a pneumatic starter.

Landing gear, flaps, bomb doors, control servos, and similar intermittent actuations are handled by high-pressure actuators, either hydraulic or pneumatic.

The other accessory described is an antiwheel-slide device called a "Decelostat." The device is an adaptation of those used on railroad cars.

An energy wheel synchronizes with the landing wheel. Wheel slip is detected instantly by overtravel of the energy wheel. Overtravel results in a momentary reduction in braking force, which averts skid.

Besides eliminating skids and tire blowout, the device aids steering and shortens stops, it was reported.

Based on discussions and six papers presented at two sessions under the chairmanships of **G. W. Newton** and **A. L. Klein.** Papers "Sand and Dust Erosion in Aircraft Gas Turbines," **J. E. DeRemer,** Air Materiel Command. . . . "New Approach to Axial Compressor Cascade Testing Technique," **J. R. Erwin** and **J. C. Emery,** National Advisory Committee for Aeronautics. . . . "Practical Conclusions on Gas Turbine Spray Nozzles," **D. R. Ganger** and **F. C. Mock,** Bendix Products Division, Bendix Aviation Corp. . . . "Auxiliary Gas Turbines for Pneumatic Power in Aircraft Applications," **H. J. Wood** and **F. Dallenbach,** AiResearch Manufacturing Co. . . . "Aircraft Decelostat—A Device for Wheel Slide Protection," **A. J. Bent,** Westinghouse Air Brake Co. . . . "History, Characteristics, Service Experience and Future Prospects of ZK60 Magnesium Extrusion Alloy," **E. H. Schuette,** Dow Chemical Co. . . . All of these papers will appear in abridged or digest form in forthcoming issues of the SAE Journal, and those approved by Readers Committees will be published in full in SAE Quarterly Transactions.

About

ORMOND E. HUNT, General Motors' executive vice-president, retired on Oct. 1. He continues as a member of the Corporation's Board of Directors. On Sept. 22 he received the Second Annual Award of Merit from the American Association of Motor Vehicle Administrators "in recognition and grateful appreciation of his immeasurable contribution to the education, economy, and pleasure of America through his activities and cooperation in development of automotive transportation." Prefaced only by a brief two years in the construction business immediately after graduating from University of Michigan, Hunt entered automotive engineering and stayed there. World War I took him from a chief-engineer post at Packard to be chief engineer in charge of Liberty aircraft engine design and production for Uncle Sam. He returned to Packard after the war, spent a year with Hare's Motors in New York and came to GM as chief engineer of Chevrolet in 1921. He went on to become a GM vice-president in charge of technical activities, a member of the board of directors and of the Operations Policy and Administration Committees, and executive vice-president. . . . He regularly contributes potently to the technical and general progress of the industry as leader or active counselor on projects of general industry concern—often in a manner which keeps his own efforts in the background and leaves to others the public acclaim.

KINDL

DEAN

OSBORN

NEWILL

CARL H. KINDL and **HUGH DEAN** have been elected vice-presidents of General Motors Corp. Kindl, who has been assistant to O. E. Hunt, will have jurisdiction over the Overseas and Canadian Group, and Dean, formerly general manufacturing manager of the Chevrolet Motor Division, will be in charge of the manufacturing staff. Kindl and Vice-Presidents **CYRUS R. OSBORN**, general manager of the Electro-Motive Division, and **EDWARD B. NEWILL**, general manager of Allison Division, were elected to membership on the corporation's administration committee. **R. M. KYES**, formerly executive in charge of the former procurement and schedules staff, has been appointed assistant general manager of the GMC Truck & Coach Division, Pontiac, Mich.

KYES

CHARLES O. BIRD is now associated with the Tarrant Mfg. Co., Saratoga Springs, N. Y., in the capacity of chief engineer.

VERNON M. ZWICKER has become project engineer at the Allison Division of General Motors Corp. in Indianapolis, Ind. He had been connected with Wright Aeronautical Corp.

FRED R. HILSON has become general manager of National Automotive Parts, Ltd., Toronto, Ont., Can.

HOWARD F. DOLL recently became assistant to the plant manager at the Lamb Electric Co. in Kent, Ohio.

WILLIAM L. ARMSTRONG has become sales manager at Ben Avon Motor Sales, Inc. in Pittsburgh, Pa.

S. W. STEININGER has become design engineer for Fairchild Engine & Airplane Corp., Oak Ridge, Tenn. He previously held a similar position with the Hercules Motors Corp., Canton, Ohio.

RALPH R. TEETOR, president of Perfect Circle Corp., Hagerstown, Ind., has been chosen as a member of the Board of Trustees of Earlham College, Richmond, Ind. It is the first time in the 100 years of Earlhams' existence that a man who was not a Quaker or Earlham alumnus was selected as a Trustee.

DR. E. A. WATSON has been elected chairman of the Automobile Division of the Institution of Mechanical Engineers for the session 1949–1950. He is associated with Joseph Lucas, Ltd., Birmingham, England, as technical director.

JOE WALKER MORLEDGE is now an automotive engineer with the Gulf Oil Corp, in Houston, Tex.

STEPHEN M. BATORI, well-known Seattle consulting engineer, recently announced the formation of Stephen Batori & Co., engineers and consultants, with offices at 200 James St., Seattle. Batori was formerly associated with DeWitt C. Griffin & Associates.

C. H. LINN has been appointed vice-president of the Union Petroleum Co. in Council Bluffs, Iowa. This company is engaged in the manufacture and sales of lubricants at wholesale. Linn's responsibility will be to head the sales and technical departments.

GEORGE C. FOWLER is now service manager and shop foreman at the Westminster Auto Service Co., Westminster, S. C.

KENNETH W. CUNNINGHAM, JR., is an industrial sales engineer for the Sun Oil Co. in Akron, Ohio.

LESTER A. LANNING has been appointed acting general purchasing agent at the New Departure Division of General Motors, Bristol, Conn. He has been affiliated with New Departure for 30 years.

STANLEY L. COHEN is a draftsman at H. Sand & Co., Inc., New York City.

Members

MANUEL PLOTKIN is a junior engineer for the Michigan Consolidated Gas Co. in Detroit, Mich.

JOHN R. MORTON has become a laboratory engineer at Refrigeration Engineering, Inc., Los Angeles, Calif.

ROY T. O'NEIL is now an engineer at Seattle Gas Co., Seattle, Wash.

JOHN J. PRENDERGAST, JR., is a powerplant expediter for the Chase Aircraft Co. in West Trenton, N. J.

JAMES C. BASH is now junior test engineer at Cummins Engine Co. in Columbus, Ind.

WILLIAM GEORGE ESTER is now a process engineer in the Rochester Products Division of General Motors Corp., Rochester, N. Y.

RUSSELL W. MEALS has become an automotive engineer at the Sinclair Refining Co. in Kansas City, Mo. He had been service manager for P. E. West Motors, Inc., Springfield, Ohio.

ROLAND F. HORTON is now sales engineer for Midland Screw Corp., Chicago. Prior to this he held a similar position with Thompson-Bremer & Co., same city.

J. H. DOOLITTLE, vice-president of Shell Oil Co., New York City, has announced a program to study performance of jet fuels and lubricants in actual flight operation, in the first jet-equipped research plane to be operated by any oil company. The plane was recently acquired from the Air Force by Shell and is now in Oakland, Calif., being fitted with analytical and recording instruments. A B-26 type bomber equipped both with twin reciprocating engines and with a jet engine, it is officially designated an XB-26F because of its experimental nature.

FRANCIS C. FLECK is now a draftsman at the General Motors Truck & Coach Division, Pontiac, Mich.

WALLACE A. CRISMORE is now manager of Mueller White Truck Co., Inc., Huntington, W. Va. He was previously a salesman for Mack International Motor Truck Corp., Pittsburgh, Pa.

F. F. MUSGRAVE is managing director of Anglamol, Ltd., London, England. Anglamol is the British affiliate of Lubrizol Corp., supplying the United Kingdom with Lubrizol additives as made in England. Musgrave had been technical assistant to the president of Lubrizol Corp. in Cleveland.

JULES LOUIS DUSSOURD recently became an aerodynamicist for the Fort Worth Division of Consolidated Vultee Aircraft Corp. He was previously an instructor in mechanical engineering at the College of the City of New York.

JULES P. KOVACS has been elected vice-president in charge of engineering of Purolator Products, Inc., Newark, N. J. He joined Purolator as a design engineer in 1929 and was appointed chief engineer in 1941. Kovacs is a member of SAE Committee A-6, Aircraft Hydraulic and Pneumatic Equipment, and is active in other hydraulic standardization groups.

JOHN S. CONANT, right, is now president of Technical Managers, Inc., New York City, **JAMES D. MOONEY**, chairman, announced. The organization provides industrial counsel and management service in production, inventory and cost controls, purchasing and materials supply methods, sales and distribution, and market forecasting and analysis. Conant was formerly general purchasing agent, and, later, director of procurement for Willys-Overland Motors, Inc., Toledo, Ohio.

RAY L. MORRISON has been elected to the newly-created post of executive vice-president of the DeVilbiss Co. in Toledo, Ohio. He has been general manager of the Brake Division of Timken-Detroit Axle Co., and previous to that was vice-president and general manager of Bendix-Westinghouse in Pittsburgh and Elyria.

GEORGE T. CHRISTOPHER, who shelved his engineering degree to start at the bottom of the industry and who has been president of Packard Motor Car Co. since 1942, has resigned to make his second attempt to become a farmer. He had been an executive of General Motors Divisions for 15 years when in 1934, as vice-president of manufacturing for Buick he "retired" to farm. Seventy days later he was appointed vice-president of manufacturing for Packard, and during World War II in charge of that company's production of 56,000 Rolls-Royce aircraft engines and 13,000 high-speed marine engines for PT boats. His 750 acre farm is near Tipp City, Ohio.

FRANK W. CALDWELL, right, receiving the 133rd John Scott Award for having pioneered the development of the controllable pitch propeller. The award is sponsored by the City of Philadelphia as a result of a fund established in 1816 by the Scotch chemist, John Scott, the income of which was to be distributed among ingenious men and women who make useful inventions. Caldwell, director of research of United Aircraft Corp., won the Collier Trophy in 1933 and the Sylvanus A. Reed Award in 1935. Presenting the award, which was accompanied by a cash award of $1000, is President Frederic D. Garman of the Philadelphia City Council, representing the City's Board of Trusts. Dr. Merle M. Odgers, president of Girard College, where the award was presented, is looking on.

EMIL O. WIRTH, formerly chief engineer of Stromberg Automotive and Light Aircraft Carburetors Department is on special assignment on the general manager's staff of Bendix Products Division, South Bend, Ind. For the past year he was assigned to budget, cost and personnel control of the Automotive Division and has recently been transferred to perform similar duties in the Aircraft Division. He is past-chairman of the SAE Chicago Section and was active in establishing the South Bend Division in 1942.

HARRY G. BOLTON has been appointed chief engineer of the Marvel-Schebler Carbureter Division of Borg-Warner Corp. in Flint, Mich. He has been associated with the Engineering Department of the division since 1928. In 1940 he was appointed assistant chief engineer.

LLOYD WOLF has been appointed chief engineer in charge of the Twin Disc Clutch Co.'s Engineering Department at Racine, Wis. He joined Twin Disc in 1947 as chief development engineer. During the war, Wolf was chief engineer of Army Ground Forces, Board No. 2, Fort Knox, Ky., where he was primarily concerned with the development of hydraulic power transmissions for the Army's heavy tank program.

CHARLES E. STEVENS, JR., is now plant manager of the Chicago Railway Equipment Co. in Chicago. He had been chief engineer of the Al-Fin Division of Fairchild Engine & Airplane Corp., Farmingdale, N. Y. The Chicago Railway Equipment Co. is a licensee of the Fairchild (Al-Fin process) and Stevens will continue the development program on bimetallic brake drums and pistons in his new position. He has presented two papers on this subject before the SAE.

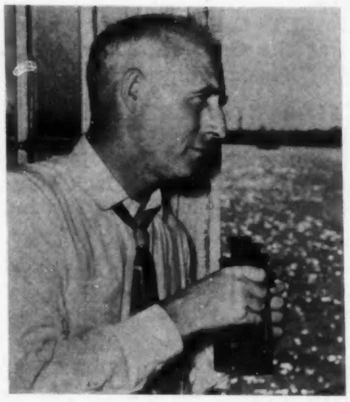
GERALD von STROH, director, Mining Development Committee, Bituminus Coal Research, Inc., addressed the American Institute of Mining and Metallurgical Engineers, in Columbus, Ohio on Sept. 27. His paper, "The Analysis for a Continuous Mining Machine," outlined, step by step, the procedure in developing a new piece of equipment. Also of particular interest was an illustrated portion of the paper on historical background of man's attempt to improve his efficiency in mining.

JESSE M. ROSEBERRY recently started his own business, Instruments Sales & Service, at 803 E. Burnside St. in Portland, Oreg. His wife is associated with him as office manager. The company will service all makes of speedometers and tachometers and will specialize in special instruments. He also will give time to plant layout work with engineers having needs in the general field of instruments. Roseberry is field editor for the SAE Oregon Section.

ROBERT D. HOLBROOK is an engineer at the Chrysler Corp. in Highland Park, Mich.

LEONARD H. KEEVE, JR., recently became production engineer for the Western Electric Co. in Chicago, Ill.

K. T. KELLER, president, Chrysler Corp.; **ALFRED REEVES**, advisory vice-president, AMA; **HARVEY S. FIRESTONE, JR.**, chairman, Firestone Tire & Rubber Co.; and **P. M. HELDT**, author and publisher of automobile engineering developments, were among those who received Distinguished Service Citations at the Automobile Old Timers 10th Anniversary on Oct. 18. Keller was the principal speaker.



THOMAS O. RICHARDS, head of the laboratory control department, Research Laboratories Division, General Motors Corp., was a member of the Selection Committee which chose the U. S. team of speed boats to compete in the 1949 British International Harmsworth Trophy event in the Detroit River

JACK C. MAILLARD is now salesman in the Used Car Department of Cadillac Motor Car Division, General Motors Corp., New York City.

FRANK NIXON has been appointed chief sales and service engineer at Rolls-Royce, Ltd., and will be at their main works at Derby, England.

DENOS A. DEMITRACK is a junior mechanical engineer for the Board of Transportation, New York City.

W. SWALLOW, director and manager of the manufacturing staff, General Motors, Ltd., London, was recently appointed to the Board of Directors of the company.

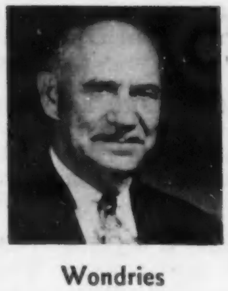
HAROLD D. KELSEY has become staff assistant to the manager of the Turbine Division at General Electric Co., Schenectady, N. Y.

LLOYD T. PETERSEN recently besame service representative for Ford Motor Co. in Dearborn, Mich.

ALEXANDER H. d'ARCAMAL was honored recently by his associates with flowers and festivities on his 30th anniversary with Pratt & Whitney Division, Niles-Bement-Pond Co. He is now vice-president and sales manager of the company's Small Tool & Gage Division, and is one of the nation's leading metallurgists. An SAE member since 1925, he has been active on various SAE technical committees.

The Tool Engineering Handbook of the American Society of Tool Engineers has just been published by the McGraw-Hill Book Co. This extensive reference of over 2000 pages covers tool design, fabrication, maintenance and utilization, product design, cost estimating, economical selection of machines, processes, and tools, and analysis and improvements of setups and operations. Among the 144 authors of the Handbook are the following SAE members: **S. P. HALL**, Design News; **H. B. CHAMBERS**, Atlas Steels, Ltd.; **W. H. OLDACRE**, D. A. Stuart Oil Co.; **JOHN SASSO**, Business Week; **S. C. MASSARI**, American Foundrymen's Association; **E. P. BLANCHARD**, The Bullard Co.; **W. F. ARDUSSI**, Variety Machine and Stamping Co.; **C. R. ALDEN**, Ex-Cell-O Corp.; **E. L. HEMINGWAY**, consultant; **G. O. HIERS**, National Lead Co.; **R. T. JONES**, Handy & Harmon; **E. BUCKINGHAM**, Massachusetts Institute of Technology; **C. H. STANARD**, Buick Motor Division, GMC; **G. PALMGREN**, SKF Industries, Inc.; **W. E. THILL**, Federal-Mogul Corp.; and **P. F. ROSSMANN**, Symington-Gould Corp.

CHARLES H. WONDRIES, since 1939 head of the National Accounts Division of the Studebaker Corp., has retired from the automobile business after a 24-year association with the company. **GEORGE E. HAMMEL**, until recently manager of Studebakers' Chicago office of the National Accounts Division, has been appointed sales manager of the division, succeeding Wondries in the South Bend post.



Wondries

Hammel

C. M. HEWITT is now vice-president of Bradley University, Peoria, Ill. He was previously dean of the University.

RALPH E. WILLIAMS is a junior mechanical engineer for the City of Detroit.

RODERICK H. CLARKE, sales engineer and wartime lieutenant colonel who helped in developing the Jeep, has been named Autocar district manager in Los Angeles with supervision over the Southern California sales area. Upon leaving the Army in 1946, Clarke joined the Autocar organization and surveyed the potential West Coast market for heavy-duty trucks. In 1947 he was transferred to their factory in Ardmore, Pa., as sales engineer.

C. FAYETTE TAYLOR, professor of automotive engineering at M.I.T., recently returned from a lecture and inspection trip to Europe, sponsored by the American-Swiss Foundation for Scientific Exchange. In Switzerland he lectured at the State Technical Institute, Zurich. He also lectured before the Society of Mechanical Engineers of Belgium in Brussels, and before a group of petroleum engineers in Delft, Holland.

TED NAGLE has been appointed executive vice-president and director of sales and advertising for the Acme-Winter Corp., Buffalo, N. Y. He was formerly public relations executive in New York, Detroit, and Boston.

E. RICHARD WALTER has been appointed mid-western sales representative of the Plastic Metals Division, National Radiator Co., Johnstown, Pa. He comes to Plastic Metals from Powdered Metal Products Corp., Chicago.

EDWARD LATTA has returned to Redmond Co., Inc., Owosso, Mich., to join the engineering staff as a special project engineer. From 1939 to 1946, he served with Redmond, during the last year as chief engineer. He then became chief engineer of Universal Electric Corp. in Owosso, where he has served until now.

ROBERT H. EATON has become a research associate at the University of Michigan, Ann Arbor.

CHAUNCEY J. HAMLIN, JR., has resigned as project engineer on rocket powerplants to join the aerophysics laboratory of North American Aviation Co., Downey, Calif.

JAMES G. BILLMEYER is a junior engineer for Bendix Products Division of Bendix Aviation Corp., South Bend, Ind.

J. A. DURR, recently resigned as technical advisor to the general manager of the Albion Malleable Iron Co. of Albion, Mich., has joined the staff of the Erie Malleable Iron Co. Erie, Pa.

DUANE E. MARQUIS is a junior engineer at Frank Wheatley Pump & Valve Manufacturers, Tulsa, Okla.

JOSEPH A. BOOTHROYD, JR., has become project engineer in the Aviation Gas Turbine Division of Westinghouse Electric Corp., Lester, Pa.

WILSON P. GREEN, professor of mechanical engineering at the Illinois Institute of Technology, has designed a double-walled engine noise test room where he can make accurate sound measurements to learn how to reduce engine noise by as much as 50%. Believed to be the only such room in existence, it is built on a four-in. platform of soft rubber and lined with a six-in. thickness of fireproof glass. Engines tested have ranged from a 20-lb, one hp lawn mower to a 200,000-lb, 2000 hp diesel.

OBITUARIES

JAMES M. SHOEMAKER

James M. Shoemaker, who was the 1947 vice-president of the Society for the Aircraft Powerplant Activity and chief engineer of Chance Vought Aircraft Division, United Aircraft Corp., died Sept. 28 in Dallas, Tex., after a long illness. He was 47.

A graduate from Purdue University in 1925, he took his master's degree at M.I.T. and joined Pratt & Whitney Aircraft in 1928, and helped to organize the Engineers' Aircraft Co. of Stamford, Conn.

He had been with the NACA on wind tunnel projects, and spent a year lecturing in aeronautical engineering at the University of Southern California.

He joined Chance Vought in 1934, became chief project engineer, and six years ago was appointed chief engineer.

TOM HUBBARD

Following a three months illness, Tom Hubbard, president of the University of Wisconsin SAE Student branch, died Sept. 10. He was 21.

He had been an honor student at the Milwaukee Shorewood high school where he was a star athlete, and had earned a major W for football in his freshman year at the University in 1945.

He had been elected to several honorary scholastic societies.

FRED W. HUBER

Fred W. Huber died Aug. 26 in Buffalo General Hospital after a long illness. He was 47.

After World War I he was with the U. S. Navy as submarine diesel engineer, and for two years was engineer on ocean vessels. His automotive experience included fleet superintendencies.

From 1935 to 1939 he was export service representative for White in South America, Asia, the Near East and Europe. At the time of his death he was field service engineer at White Motor Co., Cleveland.

CHARLES B. WHITTELSEY

Charles B. Whittelsey, who served 12 years as the treasurer of the Society, died Oct. 16 in his home in Hartford, Conn. He was 80 years old.

He began his carrer in the automotive industry in 1900 with the Hartford Rubber Works, became factory manager nine years later, was elected president in 1916 and served in that capacity for 11 years before the company became a part of the Uinted States Rubber Co. He pioneered the first single tube tires for New York City's electric buses, built the first solid tires for Mack, and equipped the famed Kitty Hawk plane of the Wright brothers with tires.

His company introduced the Dunlop straight side tires in the early days of the industry.

Whittelsey joined the SAE in 1910 and went to work at once on wheel, axle and tire standards committees, and served as councilor for two two-year terms.

In 1929 he was appointed executive vice-president of the Hartford Chamber of Commerce and 10 years later retired to become honorary vice-president of that organization. He had been president of the chamber for four years, and was a leader in bringing about the merger of the Chamber of Commerce, Board of Trade, and the Business Men's Association.

IRWIN LISS

Irwin Liss, president and general manager of Liss Aircraft Products, Inc., Silver Creek, N. Y., died last June. He was 36.

A Clevelander, he was a graduate of East Technical High School, and from 1931 to 1935 became a factory site specialist with real estate firms there. The following two years he was with Tool & Die Products, Inc., working on tool designing, and layout work on dies, jigs, and fixtures.

In 1939 he joined Thompson Aircraft Products Co. as a plant engineer and worked on tool design and manufacturing methods.

ROBERT A. WEINHARDT

Robert A. Weinhardt passed away on Sept. 26. One of the pioneer automotive engineers of the nation, who since August, 1947 had been in charge of all engine design, including the new 6-cyl engine at Willys-Overland Motors, Inc., was 66.

Taking an early interest about the prospects of the automobile, he received his education at Armour Institute of Technology in Chicago. He became an automotive engineer consultant in that city and advised numerous automotive concerns in the early days of the industry.

Weinhardt designed the Desert Flyer, an early automobile of the late sports promoter, Tex Rickard. This was a 12-passenger car designed to carry mine prospectors across the Nevada desert. The car had the first demountable disc wheels and pneumatic ride controls which later developed into the shock absorber.

He had been chief engineer of the Henry Ford Motor Car Co.; assistant chief engineer of Continental Motors, Inc.; and during World War II was a production engineer for aircraft engines at the Packard Motor Car Co. and assisted in constructing engines for PT boats.

As a Graham-Paige engineer after the war, he designed the chassis for Kaiser-Frazer automobiles and joined Willys in 1947 as automotive powerplant engineer.

G. A. KRAUS

A veteran sparkplug and automotive electrical equipment engineer, G. A. Kraus was killed in an automobile accident, June 24, at the age of 60.

He started his engineering career in 1911 with the Jeffery Dewitt Co., Detroit, following three years in mechanical and electrical engineering at Michigan Agricultural College.

In 1915 he joined Champion Spark Plug Co., Toledo, and moved to Chicago as sales engineer for that company. In 1922 he was appointed district sales manager for the central states.

Air Cargo Program

Hotel Statler — New York City

Nov. 29, 1949

(Part of the Annual Meeting of the American Society of Mechanical Engineers)

Co-Sponsored by: American Society of Mechanical Engineers
Institute of Aeronautical Sciences
and
Society of Automotive Engineers

9:30 A.M. to 12:00 NOON—TECHNICAL PAPERS

"Air Cargo Today—Here to Stay"

—Charles Froesch, Eastern Air Lines, Inc.

"What Should Be Done to Improve Cargo Aircraft"

—W. W. Davies, United Air Lines, Inc.

"Improvements Required in Air Cargo Ground Handling"

—R. Dixon Speas, American Airlines, Inc.

2:15 P.M.—INSPECTION TRIP

to Newark Airport—Foremost Commercial Air Cargo Terminal in the World

8:15 P.M.—TECHNICAL PAPERS

"Experience and Future Requirements of Military Air Cargo"

—Major-Gen. William H. Tunner, Military Air Transport Services

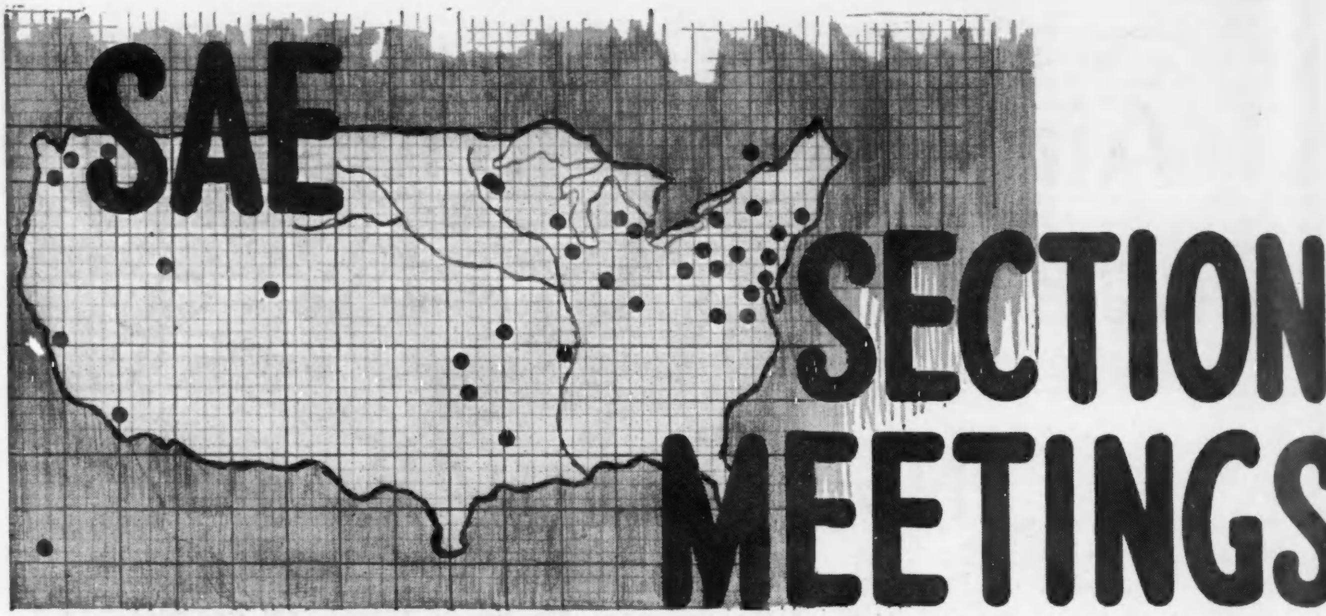
"Planning the Air Cargo Terminal"

—R. L. Hackney, Lockheed Aircraft Corp.

12:30 P.M.—LUNCHEON

Speaker: Hugh L. Dryden
Director of Aeronautical Research, NACA

Display of Cargo Plane Models and Cargo Terminal on Exhibit in foyer of Keystone Room for entire week of ASME Meeting, Nov. 28-Dec. 2.



Solvent Extraction Led to Oil Additives

• New England Section
A. R. Okuro, Field Editor

Oct. 11—Purpose of lubricating oil additives is to influence favorably the oil's effect on engine performance and deposit formation, **W. A. Howe** explained.

Along in the middle 1930's, he said, oils produced by conventional refining methods failed to satisfy certain engines in certain types of service. They tended to form sludge.

Solvent-extraction refining solved the sludge problem but increased the acid- and varnish-forming tendencies of oils. Bearing corrosion, piston seizure, and ring sticking became serious problems.

Additives were the solution that petroleum chemists and engineers found, Howe said. He is with the Gulf Oil Corp.

War Born Bimetallic Process Is Widening

• Metropolitan Section
John D. Waugh, Field Editor

Oct. 4—Encouraged by the success of the steel aircraft engine cylinder barrel used during World War II, engineers have developed designs for cast-iron cylinder ring bands to aluminum pistons, bimetallic timing gears, aluminum-steel bearings, bimetallic housings, cast-iron brake drums with bonded aluminum fins for cooling, and stainless steel-aluminum cooking utensils, **Charles E. Stevens,** chief engineer of the Al-Fin Division, Fairchild Engine & Airplane Corp., told the regular meeting of the Section this evening.

In making the cylinder, a suitably cleaned ferrous liner is immersed in a bath of molten aluminum. When the liner has reached the temperature of the aluminum, it is chemically attacked by the light alloy and an aluminum-rich alloy forms on the face.

The cylinder is removed from the molten aluminum bath, placed in a mold, and the aluminum casting is poured about it. It is possible to save machining operations when fins of sufficient thinness are cast on cylinder barrels. An improvement of 50% in heat dissipation and decrease of 25% in weight have been achieved over all-steel construction.

The bond is chemical or molecular, transfers heat from one metal to the other without loss at the interface, and also transmits stress. It is strong and runs up to Vickers 875 diamond Brinell hardness. Bond strengths have reached 17,000 psi and shear 8000 psi on tests.

Of widest automotive use to date are the bimetallic pistons where high resistance to wear and high temperature operations of a cast-iron ring band is combined with the lightness of aluminum.

Eighty percent of the wear on piston ring grooves is in the vicinity of the top ring. The cast iron band slows down wear, and the aluminum piston has a lower inertia factor than an all-ferrous piston.

There are, the speaker reported, several hundreds of such bonded bimetallic pistons in use in this country ranging in size from 9 x 14 in. locomotive diesel engine pistons to 4 x 5 in. pistons for truck, tractor, and bus use.

In one diesel engine installation, where the bimetallic piston is used, the 200,000 mile mark has been reached whereas 25,000 miles had been the life expectancy of the all-aluminum pistons heretofore used.

Bonded aluminum bimetallic timing gears are in use by the thousands. They are comparable to run-of-production fiber gears for silence in operation, and have at least three times the potential wear life and strength, he said.

The bonded cast iron and aluminum brake drum is more similar to the cylinder barrel. Properly bonded by this process, the brake drum permits fade-free operation for the first time, and actually increased liner life by 12 times.

Omitting Top Ring Saves Oil, Breakage

• Oregon Section
Jesse M. Roseberry, Field Editor

Sept. 27—The greater performance and economy of new engines, goal of automotive engineers to meet consumer demands, was brought out by SAE President **Stanwood W. Sparrow,** who presented "My Friend, The Engine" to a capacity audience.

A lively discussion on piston ring wear and breakage disclosed the fact that new high compression engines having pistons without a ring in the top groove decreased oil consumption and ring breakage.

Lt. Col. Marion E. Carl, a fighter pilot of the United States Marine Corps gave a brief talk on pilots' experiences at sonic and supersonic speeds and their task of watching the increased number of instruments.

Hollister Moore, of the SAE headquarters staff, gave a short talk on section activities.

On a whirlwind trip up the West Coast and down again, SAE President Stanwood Sparrow appeared on programs of seven SAE Sections and Groups—Spokane-Intermountain, Northwest, British Columbia, Oregon, Northern California, Fresno Division of Northern California, and San Diego—all between Sept. 22 and Oct. 3. Then he was on hand for the SAE West Coast Aeronautic Meeting in Los Angeles, Oct. 5-8.

Here are pictures taken on the tour. . . . Northern California Section's speakers table included (left to right) George Neely, Sparrow, Harry Taylor, Hollister Moore, and U. A. Patchett. . . . Sparrow saw Fishermen's Wharf in San Francisco with Northern California Section Chairman Harry Taylor. . . . Oregon Section Chairman Floyd Chapman arranged a motor trip to Bonneville Dam via the Oregon bank of the Columbia River. The party toured the dam area, inspected the power house, motored across the dam, then proceeded along the Washington bank to Vancouver and Portland. Photographed as they were admiring Multnomah Falls at Columbia River Gorge are (left to right) Chapman, Sparrow, E. A. Haas, Clarence Bear, and Henry Muessig. Sparrow also saw Grand Coulee Dam with Spokane-Intermountain Section Chairman Louis Johnson. . . . Just before Sparrow talked to almost 200 at Northwest Section's meeting, he and Section Chairman Paul Olson (center), enjoyed a word of advice from Hollister Moore. SAE Staff Man Moore accompanied Sparrow

• British Columbia Section
J. B. Tompkins, Field Editor

Sept. 26—Largest turnout ever to attend a Vancouver SAE meet was on hand to hear **SAE President Stanwood W. Sparrow.**

The Studebaker executive delivered an instructive and absorbing talk on the topic "My Friend the Engine", after commenting on the local body's increased stature, from Group to Section, with the remark "from the number attending this meeting, I'd say the alteration in status is overdue." He thanked members for their presentation to him of a minature British

Columbia Coastal Indian totem pole, hand carved by Indian craftsmen out of native north-coast slate.

Chairman H. L. Hinchcliffe reassured local members that no discount would be made from their Canadian funds if used to pay SAE dues, despite the Dominion's recent dollar devaluation.

Sparrow told of the pride which factory engineers take in hitting exacting test-model specifications "squarely on the nose." Machine-produced engines can't duplicate such exacting tolerances, he explained, so that allowance must be made, when considering the performance of test-models, for the tolerance range which will inevitably be present in specimen production units. The President told his Vancouver listeners that the trick is to so design an engine that it will operate efficiently within the relative extremes of tolerances which machine-produced units will possess.

There's a knotty problem in the designing of almost any component part of an engine, however simple, said Sparrow, illustrating his point with the comment that even fans, if left to their own devices, would operate with annoying growls, groans, or squeals. Designing the part is never the end of the story, he explained, saying that from there engineers moved on to the problems of "bugs" and how to overcome them.

Greenshields Tells How He Gets 150 MPG

• Central Illinois Section
I. R. Lamport, Field Editor

Oct. 5—A lesson in how to get 150 miles to a gallon of gasoline, taught by **R. J. Greenshields**, drew the largest and most enthusiastic crowd of this section's history. It isn't easy, they learned.

As a hobby, engineers and chemists of the Wood River Research Laboratory of the Shell Oil Co. at Wood River, Ill., work hard all year preparing their cars for the annual Mileage Marathon, explained Greenshields, director of research for Shell at Wood River.

Mileages in excess of 150 mpg have been obtained, he said, but only after a great deal of time and effort had been devoted to altering the vehicles and to developing a driving technique.

Alterations to Greenshields' car (a 1947 Studebaker) included such things as removing nearly all rubber tread from oversize tires except a narrow smooth band about ¼-in. thick in the center of the tire, and running tires at 110 psi tire pressure. A different rear axle ratio, SAE 10 oil in transmission and differential, removal of oil seals, removal of grease from wheel bearings, and lubricating with SAE 30 oil were other changes. Compression ratio was increased from 6.5:1 to 10:1. Medium cold plugs with a heavy-duty coil and a special wiring harness were also used. The fan, water pump, and generators were also disconnected. These alterations gave a maximum of about 54 mpg at a constant road speed of about 20 mph.

The additional mileage necessary to increase the 54 mpg to 150 mpg was gained by development of a special driving technique. This consisted of accelerating at full throttle in top gear from 5 mph to about 20 mph. The engine was then stopped and the car allowed to coast down to a speed of 5 mph, where the engine was restarted and the cycle repeated.

The 1947 Studebaker which won the last contest had friction reduced to such a point, he said, that it could be pushed easily on a level highway by pressing on the car with one thumb.

The present day automobile is a very good compromise for the best overall performance of the vehicle. Greenshields pointed out that greater economy in every day driving may easily be obtained by the average motorist if he pays attention to a few details. These are good mechanical condition of the vehicle; proper adjustment of the carburetor, spark plugs, and distributor; and proper lubrication; and, what is most important, reduced driving speeds.

Technical chairman for the evening was J. M. Davies, director of research at Caterpillar Tractor Co. The new section chairman, Paul Benner, presented last year's section chairman, Russ Williams, with a certificate of appreciation in recognition of past service. It was noted that among the guests were about 20 engineering students from Bradley University. Benner welcomed them to this meeting and extended an invitation to attend all subsequent meetings.

Better Engine Design Must Lick Detonation

• Philadelphia Section
G. B. Calkins, Field Editor

Oct. 12—Important changes in automotive engines are imminent and desirable as well as economically feasible, declared **Alex Taub** in his paper, "Mechanical Octanes," in which he explored the potentialities of antidetonation design in internal-combustion engines.

The fuel, he said, is already the best designed component of the powerplant. Further major improvements in antiknock quality cannot be expected because of the cost involved.

But higher compression ratios are a source of potential savings to the consumer in miles per gallon. To get them, detonation problems must be licked—not by waiting for a hypothetical superfuel, but through improved engine design.

Taub, an automotive engineer with over 20 years of experience, described several "broadminded" engines capable of using 70 octane number fuel up to 11:1 compression ratio.

These included slide-valve, rotating

Shown at dinner preceding Canadian Section's October meeting at Ancaster are (left to right) Norman G. Shidle, Section Chairman W. W. Taylor, Vice-Chairman Col. Malcolm P. Jolley, and Vice-Chairman D. C. Gaskin. Shidle, SAE Journal's executive editor, gave his observations on "Years and Years with Engineers"

combustion chamber, and single-valve types which, he indicated, have been under development for a number of years.

Other methods of saving fuel were also discussed, including the burning of leaner mixtures at part throttle, up to 19:1 air-fuel ratio. Mixture distribution and spark plug location, as well as fuel "burnability," were named as limiting factors in this undertaking.

In the discussion, it was pointed out that combustion shock at high compression ratios must be dealt with, even if knock is eliminated by fuel chemistry; hence, design improvements remain on the engineer's agenda in any case.

One approach to a purely engine-design solution, it was recalled, was a combustion chamber in which fuel and air were kept separate until ignition, thereby preventing formation of the offending end-gas.

Expects Small Planes to Yield to Helicopter

• Williamsport Section
George Hoover, Field Editor

Sept. 12—"Helicopters are definitely here to stay and have proved their worth to the Army and the Navy," asserted **F. N. Piasecki** of the Piasecki Helicopter Corp. He further ventured the opinion that helicopters will soon supplant the short range conventional aircraft, especially among private owners.

Piasecki admitted, however, that there is much to be done by mechanical engineers to refine the design of transmissions, clutches, and drive couplings. He said that helicopter fuselage design could be improved if powerplants designed specifically for helicopters were available, instead of the present conventional aircraft engine adaptations.

Piasecki discussed his paper dealing with the history of the helicopter and its design problems. He showed two films, one dealing with the use of Piasecki helicopters replacing assault boats in marine beach landing maneuvers and the other dealing with preliminary acceptance tests for the Navy Bureau of Aeronautics. The Section was very much surprised at the performance and maneuvers of these Piasecki tandem rotor type helicopters.

In discussion, blade icing was admitted to be a serious problem. Conventional aircraft wing deicing methods upset the delicate balance of the lift and load characteristics of the rotor blades.

Melting the ice off the leading edge results in the water refreezing on the trailing edge, and the rubber boot method destroys the blade section to such an extent that the blades lose their lift properties. Now being tried

You'll Be Interested to Know . . .

AT THE 20TH ANNUAL SAFETY CONVENTION AND EXPOSITION in New York on March 28-31, 1950, Herbert Happersberg, Brockway Truck Co., will be the official SAE representative.

SAE-NOMINATED DIRECTORS OF CRC for the two-year term beginning Jan. 1, 1950, will be R. D. Kelly, Arthur Nutt, E. N. Cole, and G. J. Huebner, Jr. (Cole and Huebner will be serving their first terms as SAE-nominated directors of the Coordinating Research Council.) Other SAE directors, whose terms still have one year to run, are C. E. Frudden, E. S. MacPherson, and R. L. Weider.

INDIANA STATE TECHNICAL COLLEGE now has an SAE Student Branch. The Council made it official on Sept. 15. Faculty adviser

Left to right: Warren D. Berkley, Dr. Ivan Planck, William McCormick, and College President Archie T. Keene

for the new Branch is Dr. Ivan Planck, head of the Department of Mechanical Engineering, and William McCormick of Pittsburgh, Calif., is Student chairman. Representing the SAE Council, Warren C. Berkley, American Steel Dredge Co., presented the charter officially.

Admiring California State Polytechnic College's new SAE Student Branch charter are (left to right) Tom Hardgrove, faculty adviser; Southern California Section Chairman Reagan Stunkel, who presented the charter; Alvin Gorenbein, Student Branch chairman; and Ed Rentz, manager, SAE West Coast office. Stunkel, president of Aviation Maintenance Corp., told the branch's 68 enrollees that he prefers to hire engineers having practical as well as theoretical training

ENGINEERS working for N. V. de Bataafsche Petroleum Maatschappij, Handelszaken, Batavia, Indonesia, are eager to get the SAE Journal in a hurry. . . . eager enough, that is, to pay 104 U. S. dollars for air mail postage alone for the 12 issues, to be sure to get them half way around the world as fast as man's ingenuity permits. Air mail stamps total nearly $9 an issue to make the 9621-mile run from Lancaster, Pa., where the Journal is printed. Nonmember subscription is a mere $10 in the $114 total.

Proud occasion for new SAE Atlanta Group is presentation of charter by SAE Vice-President Max M. Roensch (left) to Past-Chairman Randolph Whitfield (right). Chairman John Rogers watches. Roensch conveyed a cordial welcome from Pres. Sparrow and the SAE Council to the new group

is removal of ice in sections by high intermittent heat which only loosens the ice; thus there is no water to refreeze.

In appreciation of Al Creighton's service as 1948-1949 Section chairman, a Certificate of Award was presented to him by Chairman Ingram before introducing the speaker.

Fuel Ratings May Rise 3 Octanes in 3 Years

• Kansas City Section
K. J. Holloway, Field Editor

Oct. 11—The problems of engines, fuels, and lubricants are inseparable, **Max M. Roensch**, research coordinator for the Ethyl Corp. and SAE vice-president pointed out in discussing engine design for past, present, and future fuels.

The petroleum engineer is striving for gasoline of (1) improved anti-knock quality, (2) proper volatility, (3) high purity, (4) satisfactory cleanliness, and (5) low cost, while the automotive engineer is striving for engines of (1) increased performance, (2) improved economy, (3) greater durability, (4) lighter weight, (5) less noise and vibration, and (6) low cost. The relation of these problems to each other necessitates coordination between the petroleum and the automotive industries, Roensch said.

One of the most important of the gradual changes in fuels, he continued, is the increase in anti-knock quality. The Motor Method octane number of regular gasoline has increased from approximately 60 in 1930 to 77 in 1949, and indications are that it will reach 80 in 1952. The average premium grade gasoline has also shown an increase from approximately 70 in 1930 to 81 in 1949, and it is estimated that it will be 84 by 1952.

The compression ratio of the average automobile engine and the anti-knock quality of available fuels are closely related. The compression ratio of the average passenger car engine has gradually increased from 4.4 in 1925 to 6.8 in 1949, thus demonstrating coordination between petroleum and automotive engineers.

If engines progress along the lines of presently known designs, fuel of approximately 100 octane will be required for engines of 11:1 compression ratio.

Analysis of data available has shown that appreciable increase in fuel economy can be obtained by increasing the compression ratio of passenger car engines. Roensch indicated that new engines designed for a compression ratio of 8.5:1 and upward may give as much as 28% more miles per gallon with no sacrifice in performance.

With regard to truck and bus engines, data have revealed that a noticeable increase in horsepower may be obtained by increasing the compression ratio. However, data have also indicated that the gain in output which may be obtained by supercharging is many times that obtained from increasing the compression ratio. In this regard, Roensch indicated that an important point to note is that there is no appreciable change in brake specific fuel consumption with the higher horsepower obtained by supercharging.

Experience has shown that when engine output is increased either by increasing the compression ratio or the volumetric efficiency (supercharging), modifications to the engine are usually necessary to maintain durability at the previous level.

Difficulties are encountered with pistons, rings, valves, engine cooling, and preignition, as a result of increased output. Design changes are necessary to eliminate these difficulties.

Antarctic Expedition Described at Hawaii

• Hawaii Section
Rene Guillou, Field Editor

Sept. 19—Although a diesel-electric "snow cruiser" failed the U. S. Antarctic Service Expedition of 1939-1941, an ancient Condor airplane saved the explorers, Hawaii's automotive engineers were gratified to learn from **Richard B. Black** and his movies of the expedition.

Black was in command of the East Base of the expedition, directly south of Cape Horn, and is now first assistant director of the Hawaii Aeronautics Commission.

The members found cold blue water, white icebergs, drifting snow, and sledge dogs a striking contrast to their own tropical paradise. On the professional side, Block described the pioneer work of the Expedition in operation of automotive equipment under antarctic conditions. A tractor was operated successfully at -77F. Dogs and men worked at -55F, although internal forstbite was considered a serious hazard at temperatures below -45F.

Unloading and early progress of a huge diesel-electric "snow-cruiser" was shown in the movies. Designed to carry fuel and supplies for itself and crew for a 1500-mile cruising range, this device was provided with individual electric motors in each of its four wheels, and was expected to travel over lightly packed snow on low-pressure pneumatic tires.

The antarctic icecap provided insufficient flotation and the cruiser was finally abandoned four miles from the landing place. Dog teams proved most successful for surface exploration.

The one plane at the East Base, a 10-year old Condor with low landing and take-off speeds, was repeatedly damaged by treacherous surface conditions and each time repaired.

It was used for reconnaissance of much previously unknown territory and finally transported the entire personnel of the base to an embarkation point and safety in the antarctic fall of 1941, after their original landing place had been blocked throughout the summer by unusual ice conditions.

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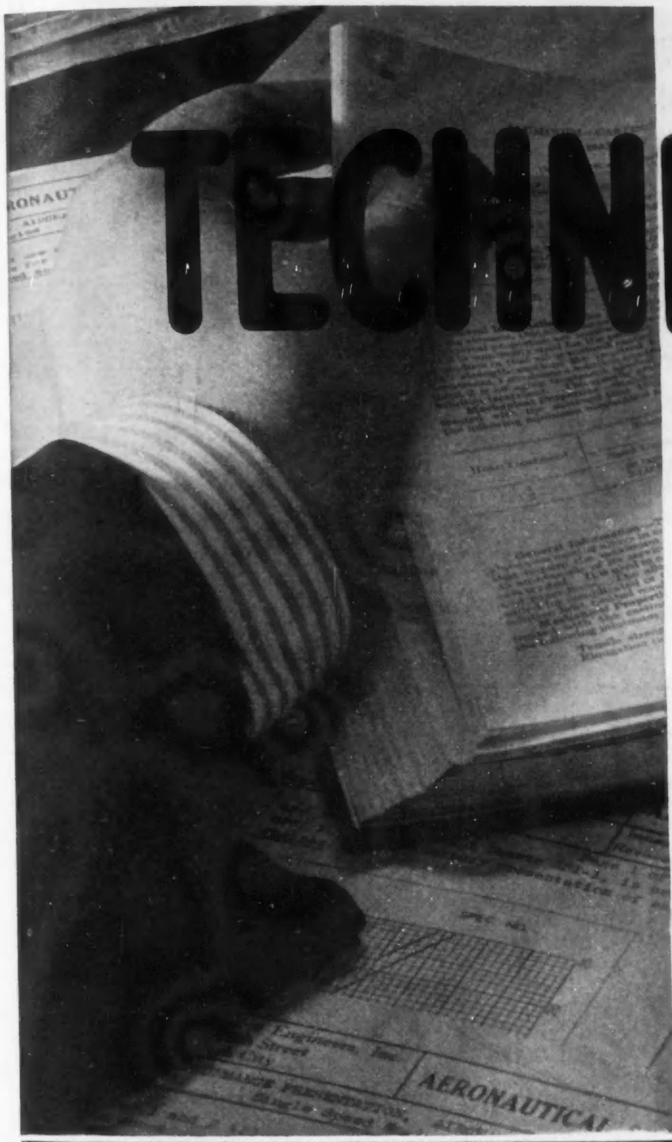
TECHNICAL COMMITTEE PROGRESS



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New Spring Reports Cover Hot and Cold Coiled Types

TWO new reports on helical coil springs, developed by the SAE Spring Committee, recently were approved by the SAE Technical Board for Publication in the 1950 SAE Handbook.

Adopted as SAE Recommended Practices, these reports are "Helical Hot Coiled Compression Springs for General Automotive Use" and "Helical Cold Coiled Compression and Extension Springs for General Automotive Use."

Both recommended practices furnish a system of dimensional tolerances which inform the user of practical manufacturing limits. The reports point out that closer tolerances are obtainable where accuracy is required and increased cost justified (valve springs, for example), and that springs can be made cheaper by omitting some tolerances.

The report on hot-coiled springs covers those made from round steel bars $\frac{3}{8}$ in. in diameter or larger. Ends of such springs are closed and squared, the bar ends usually being forged or taper rolled. Also included is a list of definitions for terms such as solid height, free height, loaded height, and nominal solid load. Helical springs for motor vehicle suspensions are not included in this report since they are covered by an earlier SAE Recommended Practice, Design and Application of Helical and Spiral Springs.

The new recommended practice on cold coiled springs covers those made from round steel wire $\frac{3}{8}$ in. in diameter or smaller.

Members of the SAE Spring Committee, of which R. R. Teetor, Perfect Circle Co. is Technical Board Sponsor, are: Tore Franzen, chairman, Chrysler

Corp.; H. H. Clark, vice-chairman, Spring Division, Eaton Mfg. Co.; P. E. Eckberg, American-Fort Pitt Spring Division, H. K. Porter Co.; H. J. Elmendorf, American Steel & Wire Co.; J. R. Gustafson, Ford Motor Co.; N. E. Hendrickson, consulting engineer; H. C. Keysor, American Steel Foundries; R. M. Neff, Chevrolet Motor Division, GMC; George Nelson, L. A. Young Spring & Wire Corp.; E. G. Peckham, Buick Motor Division, GMC; Robert Schilling, Research Laboratories Division, GMC; Bernhard Sterne, Chrysler Corp.; A. M. Wahl, Westinghouse Electric Corp.; T. R. Weber, Railway Steel-Spring Division, American Locomotive Co.; and F. P. Zimmerli, Barnes-Gibson-Raymond Division, Associated Spring Corp. Dr. R. Genders, Coil Spring Research Organization, England, is a consultant member of the Committee.

CRC Releases Eleven Reports

THE following Coordinating Research Council reports have been released for distribution and are available from SAE Special Publications Department, 29 West 39th Street, New York 18, N. Y. (This is a complete list of CRC reports released since publication of the listing of CRC reports on p. 26 of the March, 1949, SAE Journal.)

LUBRICANTS

Engine Oil, Motor

CRC-231—A Study of Low Temperature Characteristics of U. S. Army Heavy-Duty Oils (9/7/48) Price: $1.50 to SAE members; $3.00 to nonmembers

MOTOR FUELS

Detonation

CRC-232—Rating Small Samples of Motor Gasolines (5/21/48) Price: $1.00 to SAE members; $2.00 to nonmembers

LUBRICANTS

Engine Oil, Aviation

CRC-233—Ring-Sticking Tendencies of Aviation Engine Lubricating Oils (11/23/48) Price: $1.00 to SAE members; $2.00 to nonmembers

MOTOR AND AVIATION FUELS

Volatility

CRC-234—Application of Equilibrium Air Distillation to Gasoline Volatility Performance Calculations (9/48) Price: $.75 to SAE members; $1.50 to nonmembers

AVIATION FUELS

CRC-235—Symposium on Aviation Reciprocating Engines and their Fuels (9/10/48) Price: $1.50 to SAE members; $3.00 to nonmembers

MOTOR FUELS

Detonation

CRC-236—Octane Number Requirement Survey, 1948 (revised 2/10/49) Price: $2.00 to SAE members; $4.00 to nonmembers

Volatility

CRC-237—Vapor Lock Tests, 1948 (1/11/49); (revised 2/15/49) Price: $1.00 to SAE members; $2.00 to nonmembers

DIESEL FUELS

CRC-238—The Use of the CRC-Photovolt Smokemeter in Measuring Smoke from Diesel Engines (4/18/48) Price: $.50 to SAE members; $1.00 to nonmembers

AVIATION FUELS

Gasoline Additives

CRC-239—Second Supplement to the

Technishorts . . .

HYDRAULIC CONTROLS: Requirements for fittings, attachment flanges, and drives for hydraulic systems on earthmoving machines are being established by the Hydraulic Power Control Subcommittee of the SAE Construction and Industrial Machinery Technical Committee. Headed up by E. C. Iverson, J. D. Adams Mfg. Co., this group is studying the relationship between the needs of their machines and AN and SAE standards and Joint Industry Conference tube fitting specifications.

V-BELTS: To the SAE Standard on V-Belts and Pulleys has been added a new size with a nominal top width of 0.380 in. It is intended primarily for automotive applications, such as fan, generator, and pump drives. This new standard size for v-belts and pulley grooves was developed by the SAE-ASTM Technical Committee A on Automotive Rubber, recommended for adoption as an SAE Standard by the SAE Engine Technical Committee, and approved as such by the SAE Technical Board.

PUNCH AND DIE SETS: A proposed American Standard for Punch and Die Sets recently has been approved by SAE, one of four sponsors of ASA Sectional Committee B5, which develop this proposed standard. The other sponsors are ASME, National Machine Tool Builders' Association, and Metal Cutting Tool Institute. If and when the proposal receives approval of all four sponsors, it will be transmitted to the American Standards Association for final approval and identification as an American Standard. The proposed standard covers the design and dimensions for two-post, punch press tool die sets. Types and range of sizes included, notes the Committee, have been carefully selected with a view to meeting the largest volume of present user needs, and, therefore, may be manufactured on a continuous production basis.

HELICOPTER PERFORMANCE: SAE Aeronautical Information Report No. 27, Formulas for Determining Preliminary Performance Characteristics of the Helicopter, recently was issued. Prepared by the SAE Helicopter Committee, the report contains simple formulas for preliminary or approximate performance calculations. The Committee feels the time is not yet ripe to standardize helicopter performance calculations. It points to airplane performance history, in which complicated methods yielded to simple semi-rational and semi-empirical formulas and procedures, and hopes that helicopter performance determinations will follow a similar pattern.

1944 Desert Storage Tests on Aviation Gasolines with and without CS—18-24 months storage data—(1/11/49); (Second supplement to CRC-21) Price: $1.00 to SAE members; $2.00 to nonmembers

MOTOR FUELS

Volatility

CRC-240—Review of Vapor Lock Road Tests, Indio, California, 1946 (3/1/49); (Supplement to CRC-223) Price: $1.00 to SAE members; $2.00 to nonmembers

AVIATION FUELS

Detonation

CRC-241—Method for Expressing Aviation Fuel Antiknock Ratings (4/18/49) Price: $2.00 to SAE members; $4.00 to nonmembers

Price: 50¢ to SAE members, $1.00 to nonmembers. It is available from the SAE Special Publications Department, 29 West 39th St., New York 18, N. Y.

Report Experience With CRC Smokemeter

BEST methods of installing and operating the CRC-Photovolt Smokemeter, evolved by owners of the instrument, are summarized in the report "Use of the CRC-Photovolt Smokemeter in Measuring Smoke from Diesel Engines," recently released by the Coordinating Research Council, Inc. The Full-Scale Procedure and Instrumentation Group of the Diesel Fuels Division prepared the report.

Information about exhaust flow characteristics and design variables which affect them, given in the report, are said to be important in getting best results in the varied applications in which the Smokemeter may be used. (See Fig. 1 for description of the device.)

For example, experience has shown that particular attention must be paid to the sampling connection at the exhaust pipe to get a representative exhaust sample. Breathing of the exhaust line between exhaust impluses and complexity of gas flow through an exhaust pipe are said to be troublesome factors in this respect. Also given are ways of eliminating condensate and soot in the sampling line.

A detailed step-by-step procedure for measuring diesel engine smoke with the Smokemeter, in the report, explains two ways of taking measurments—continuous sampling and batch sampling.

While choice of method depends on the installation and the type of test, the continuous technique is said to be almost mandatory for cases where engine exhaust conditions change rapidly, as during acceleration. This method may give slightly higher smoke density readings because of the greater turbulence within the Smokemeter tube.

Instructions for care and maintenance of the Smokemeter also are given in this report.

The report, CRC-238, has six 8½ × 11-in. pages, including two drawings.

CRC Group Evolves Ring-Sticking Test

A research method for evaluating ring-sticking characteristics of aviation engine oils is disclosed in a report recently issued by the Coordinating Research Council. This test procedure was developed by the Cooperative Oil Test Engine Group Working on Aviation Projects, of the CLR Engine Oil Division.

In this method, "Research Technique for the Evaluation of Ring-Sticking Tendencies of Aviation Engine Lubricating Oil (CRC Designation L-30)," oils are evaluated by comparing the length of time required for actual ring sticking to occur, when the engine is operated under the test conditions. Ring sticking is indicated by an increase of temperature of the cylinder liner, measured near the top of ring travel.

The report states that equipment required for running this test includes a single-cylinder, spark-ignition internal combustion engine with a dual spark-plug type head (CRC Designation EL-30), together with suitable accessory equipment. Instruments and accessories required include a blowby meter and thermocouples.

The thermocouples are used to mea-

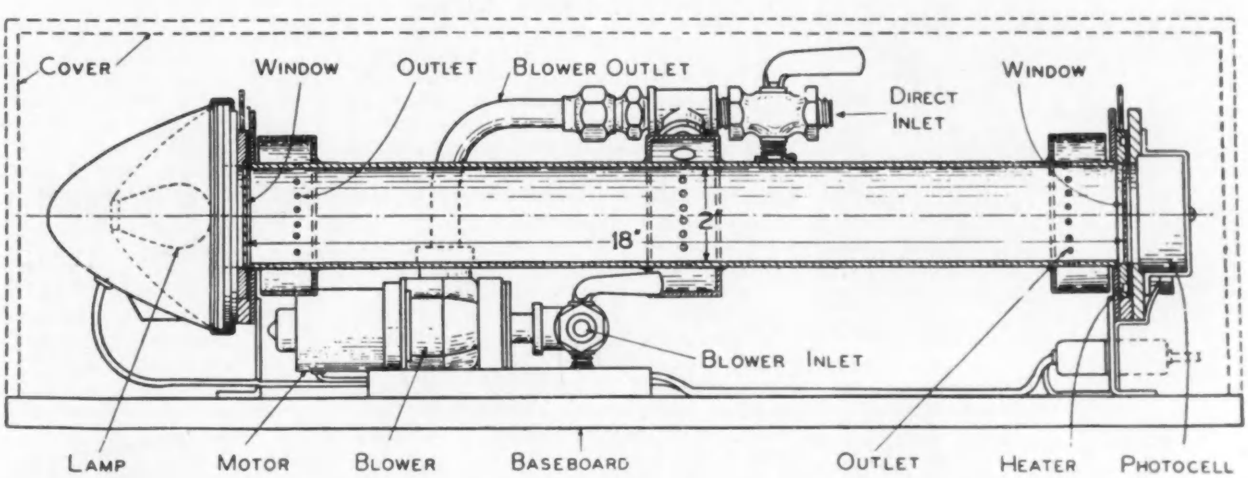

Fig. 1—The CRC-Photovolt Smokemeter consists of an 18 x 2-in. ID tube, fitted with glass windows at each end. A light source is mounted at one end, and at the other a photocell connected to a microammeter, which is graduated in percent smoke. A blower and a system of valves is provided for either introducing an exhaust sample into the tube, or for scavenging the tube with clean air.

In operation, the light intensity is adjusted to give full scale deflection (zero smoke) on the microammeter when the tube is free of smoke. Upon introduction of the exhaust sample into the tube, the meter deflection drops off in proportion to the smoke density, but percent smoke meter reading increases. One hundred percent smoke is arbitrarily defined as that amount of smoke which will just extinguish all the light inpinging on the photocell; in this case the meter deflection is zero

sure temperature of the following: cylinder-liner wall at both top and bottom of ring travel, intake air, spark plug, oil, exhaust gas, jacket coolant, and jacket outlet. The report also spells out conditions of test engine operation, inspection and maintenance of apparatus, and surface preparation of cylinder, piston, and rings.

Also detailed in the report are results of nine tests, conducted in accordance with the procedure CRC Designation L-30, at Wright Field and those from 14 tests conducted at the Research & Development Laboratories of the Pure Oil Co.

The Wright Field tests, notes the report, showed that ring sticking normally occurs in the antithrust face area of the piston. It seemed logical that a liner thermocouple located in this area would indicate more accurately the time at which ring sticking occurred than was possible by placing the thermocouple in a previously selected location. Pure Oil Co. tests substantiate this theory.

This method of indicating ring sticking—by means of a rise in top-liner temperature, measured by thermocouples at the top of the cylinder over the rear main bearing and at the center of the antithrust face—has proved to be sufficiently sensitive, accurate, and dependable, the report concludes.

The report, CRC-232, has twenty-eight 8½ × 11-in. pages, including four tables. Price: $1.00 to SAE members, $2.00 to nonmembers. It is available from the SAE Special Publications Department, 29 West 39th St., New York 18, N. Y.

New CRC Procedure Measures Vapor Lock

A technique for determining vapor lock in passenger cars, based on tests peformed in California, is presented in the report "Vapor Lock Tests—1948," recently issued by the Coordinating Research Council. It was developed by the Equipment Survey Group of the CFR Motor Fuels Division.

Called the "Tentative Parking Lot Survey Test Technique for Determination of Vapor Lock in Passenger Cars," this test procedure is considered a good, usable guide for getting reproducible test data. This method, with a few modifications outlined in the report, will permit procurement of vapor lock data on a uniform basis, the report notes.

A summary of the results obtained in the West Coast survey is given in the report. Fifteen cars, representing nine different makes and models were tested. Although the limited number of vehicles are not considered to provide conclusive data, inspection of the

results indicates a remarkable degree of reproducibility on the same vehicle, even in tests conducted at widely different atmospheric temperatures, and good agreement among various vehicles of the same make and model.

The report describes the test technique and discusses it, and analyzes data obtained from the California tests. Representatives of the following companies participated in the Pacific Coast vapor lock tests: California Research Corp., Ethyl Corp., General Petroleum Corp., Richfield Oil Corp., Shell Development Co., Union Oil Co., and Tide Water Associated Oil Co.

The report, CRC-237, has twenty seven $8\frac{1}{2} \times 11$-in. pages, including two tables, six charts, and one drawing. Price: $1.00 to SAE members, $2.00 to nonmembers. It is available from the SAE Special Publications Department, 29 West 39th St., New York 18, N. Y.

CRC Report Bares Engine-Fuel Issues

PROCEEDINGS recently were released on the Symposium on Aviation Reciprocating Engines and Their Fuels, held in Cleveland last year, under sponsorship of the CFR Aviation Fuels Division, of the Coordinating Research Council. Included in this report are seven complete papers, discussions of them, and policy statements from representatives of the military services on the piston engine's future.

The papers published in this report cover knock test technique, volatility, preignition, tetraethyl lead deposits, and fuel stability.

In his paper "Knock Testing of Fuels," E. F. Miller, Socony-Vacuum Laboratories, reports on the development of the F-21 technique, for evaluating aviation fuels under rich and lean conditions at various intake air temperatures. He also compares it with F-3 and F-4 methods.

A. L. Beall, Wright Aeronautical Corp., reports on the elementary effects of volatility, covering the pros and cons of adding heat for, and extending the time of, vaporization. He raises the question of whether to use more volatile fuels, and will these fuels boost range and reliability of aircraft powered by piston engines.

The third paper, "Preignition from Combustion Chamber Deposits," by A. Hundere, California Research Corp., reports on tests in a single-cylinder engine to study the preignition characteristics of combustion chamber deposits. Hundere shows that lead deposits are incapable of supporting preignition advanced enough to produce harmful effects. He concludes that certain fuel and oil constitutents

can promote destructive preignition.

The question of lead deposits is discussed in detail in the paper "Fuel Quality as a Restriction to Aircraft Utilization." by E. A. Droegemueller, Pratt & Whitney Aircraft. Complexities of the lead fouling problem as well as recommendations for overcoming it are spelled out here by Droegemueller.

The last three papers included in the symposium report cover fuel stability, with D. H. Moreton, Douglas Aircraft Co., Inc., reporting on "Color Stability," W. E. Kuhn; The Texas, Co., on "Induction System Deposits in Aircraft Engines," and J. S. Bogen, on "Laboratory Tests on Aircraft Induction System Deposits As Conducted by the Universal Oil Products Co."

Comments from military personnel, noted in the report, indicate that the reciprocating engine is still vital to national security. They emphasize the urgency of improving existing fuels and engines to get greater power and reliability and to decrease specific fuel consumption.

The report, CRC-235, has ninety-nine $8\frac{1}{2} \times 11$-in. pages, including 40 pages of illustrations and charts and one table. Price: $1.50 to members, $3.00 to nonmembers. It is available from SAE Special Publications Department, 29 West 39th St., New York 18, N. Y.

Section News

Cont. from p. 86

Light Plane Performs Almost Like Helicopter

• Southern New England Section
L. H. Frese, Ass't. Field Editor

Oct. 5—For light personal airplanes to be practical, **Prof. O. C. Koppen** is convinced they must fulfill two basic requirements:

1. They must be capable of being stored at the owner's home.
2. They must be useful for his commuting to and from work.

The helicopter comes closest to meeting these problems, but Koppen, who lectures on airplane design at M. I. T., is convinced that 80% of helicopter performance can be developed into a fixed wing type of aircraft. He is proceeding to demonstrate it with his K-B Helioplane.

The design specifications set up for the Helioplane required that it was to operate from a strip having a 200-ft length of usable surface and to be able to clear a 50-ft high obstacle placed at each end of a 600-ft strip. Koppen stated that to accomplish this, he knew it would be necessary to: (1) double the propeller thrust of a conventional light plane; (2) double the maximum lift coefficient; and (3) double the stalling angle of attack.

That the inventor of the Helioplane was successful in achieving his goal is evident in the performance of the K-B ship, which will take off or land fully loaded in still or gusty air, in less than 100 ft and will clear a 50-ft obstacle in less than 300 ft. It lands at 20 to 30 mph ground speed. Three developments in wing construction plus the use of a large diameter slow-speed V-belt driven propeller, and a specially constructed undercarriage are responsible for the startling performance of the Helioplane. The trailing edge of the wing has full-span pilot-actuated slotted flaps. On the leading edge is an ingenious automatically operated slat which is free to slide in or out parallel to the edge of the wing and which positions itself automatically. Built into the top surface of the wing and to the rear, is a pilot-actuated spoiler for yaw compensation. Otherwise the plane is aerodynamically similar to a conventional plane and has the cruising characteristics of a conventional plane.

An engine cooling problem which arose due to the low takeoff and landing speeds was overcome by the addition of an ejector tube to the exhaust system—an idea borrowed, Koppen stated, from the steam locomotives, which have made use of it since 1825.

The relatively slow speed propeller, 1200 rpm, reduces the noise level to that of an average automobile on the highway, an important factor since the plane is intended for use in residential areas.

Movies of repeated takeoffs and landings of the Helioplane had the audience, which contained many owners of light planes, gasping at the rate of climb in takeoff and the short distance in which the plane came to a stop at landing. The plane's angle of attack is 16 deg and its path angle at takeoff is 24 deg, making an angle of 40 deg with the ground at takeoff. There is no lifting of the tail until the nose is a good 5 ft from the ground, so marked is the rate of climb.

Chrome Plating Called Boon to Piston Rings

• Williamsport Section
George Hoover, Field Editor

Oct. 3—The chrome plating of piston rings, a wartime development, is the greatest single improvement in the

conception of the ideal piston ring in regard to long ring life, reduced bore wear, elimination of scuffing, and low oil consumption, in many years, stated **Arthur M. Brenneke,** chief engineer of the manufacturers division, Perfect Circle Corp.

It is the only type of ring available that will operate satisfactorily in bores of all standard materials and finishes, with the exception of chrome plated bores, which are not compatible, he continued. While further improvements in rings will be made, it is not likely they will be as startling as chrome plated rings.

Chrome plated rings were originally developed for aircraft engines, but the automotive industry has become aware of their advantages and is increasingly converting to their use. Chrome plating decreases the cylinder and ring wear rate by a nominal 80%.

Robert Jackson, also of Perfect Circle, presented a color film depicting the complete processing required for chrome rings from the raw material state to assembly in automotive units.

CAA More a Service Than a Police Agency

• Metropolitan Section
John D. Waugh, Field Editor

Sept. 15—The Civil Aeronautics Administration was typified as chiefly a service organization to the aircraft industry and not a "police" body by **Herbert M. Toomey** at the 1949-50 season-opening session.

Toomey, who is chief, Aircraft Division, First Region, CAA, presented a comprehensive outline of the organization and operation of the CAA and its administration of regulations and dissemination of information. The CAA was characterized as the executive branch of aviation government with approximately 14,000 employees and a 1950 budget of over 200 million dollars.

The CAA in 1948 distributed more than eight million items of aviation information and provided numerous aids to aviation manufacturers, airlines, and pilots, through ten national and territorial regions, according to Toomey. The CAA's educational role was described as an effort to "help and encourage schools to introduce aeronautical studies in all grade levels." A current CAA-college project is the development, at Texas A & M, of a fixed-wing "industrial" aircraft to be designed specifically for industrial working conditions, such as dusting.

A principal activity of the CAA is to maintain air safety, the speaker stressed. The organization builds and maintains the Federal Airways for airlines and private flying. It develops navigational aids, such as the omnidirectional range, advises on the design and construction of airports, and administers the certification of pilots, airplanes, mechanics, and air training schools.

Airframes, engines, propellers, and associated equipment are all thoroughly tested by the CAA, and advice on design is tendered where needed. Vibration, a major area of aircraft engineering investigation, is surveyed in all major aircraft components. Small manufacturers, without the means to conduct intensive vibration work, are assisted considerably. Other safety problems, such as windshield "birdproofing," fire protection, fuel and lubricant development, and a host of others are continually under study and progress.

Toomey sugested that the proposal to place a transport plane in scheduled cargo service for 9 to 12 months prior to carrying passengers is noteworthy and would help uncover inherent "bugs."

At all times the CAA assists the manufacturer and aircraft operator when difficulties do arise. A highly integrated information exchange system informs all CAA offices of what to expect in the way of malfunctioning of specific equipment, and every effort is made to head off trouble before it is repeated.

Toomey related the considerable interest of the CAA in the development and use of helicopters. He stated that his organization had taken an early interest in this field of aeronautical progress and had accumulated wide and varied experience in its design and operational activity. Vibration difficulties and fatigue life problems were under active study, and certification data were being accumulated steadily. Toomey concluded with the advice that the CAA was guided by a spirit of suggestion and cooperation and that no "shackles" of regulation will be applied to aviation other than "those which the industry itself feels unable to assume."

Allen Dallas, of the Air Transport Association of America, praised the CAA for its safety work and particularly mentioned the steps taken to prevent transport craft baggage compartment and landing brake fires. He added that the transport industry hopes that the CAA doesn't go "overboard" on safety though, because an airplane could be made so safe that it wouldn't get off the ground.

M. G. Beard, director of flight engineering for American Airlines, stated that he had watched the CAA grow from a necessity to a fact. Crediting CAA for great advances in traffic control, Beard stated that he believed traffic control now was lagging behind airframes and engine progress and unless more "brainpower" was applied to the problem, jet transports would be on the scene without adequate control for their faster travel. Beard called for a start on the work now.

John Reese, in charge of contacts with the CAA for the Propeller Division of Curtiss-Wright Corp., discussed the administration of CAA regulations as applied to propeller manufacturers. He had praise for the constructive criticism and help of the CAA and observed that while the policy of decentralization of CAA offices was frequently of material help, the great distance separating the propeller and engine manufacturing area of the East and the transport building region of the West, occasionally delayed testing work.

Chairman Richard Creter advised the membership that if a sufficient number of members wished to resume 6 p.m. dinner meetings, arrangements could be made for use of the Statler's Manhattan Room. He also advised that Miss Judy McCormick at SAE Headquarters should be contacted by all those who wish to register their desire to have the dinner meeting again.

Wrights Corrected Errors in Lift Data

• Dayton Section
T. O. Mathues, Field Editor

Oct. 4—One reason for the Wright brothers' success with heavier-than-air craft was that they discovered the inaccuracy of existing aerodynamic data and developed more reliable data with their own equipment, **Max P. Baker** showed.

Baker is special projects engineer with Inland Mfg. Division of GMC. As a hobby, he has been tracing the Wright Brothers' early notes and instruments.

He found a paper written by Wilbur Wright describing how the first airplane he and Orville built in 1900 flew

but failed to attain expected performance.

After another disappointment with a second machine, they concluded that their reference tables of lift and drag values were inaccurate.

Before the end of 1901, they had built two wind tunnels—the first out of a short wooden box for shipping starch and the second about 8 ft long—and were collecting data accurate for prediction of airplane performance.

On the same program, Capt. J. G. Goppert, of Wright-Patterson Air Force Base, described how present-day aircraft evolved from the Wright brothers' early research.

STUDENT NEWS

University of Colorado

Lubricating oil analysis can minimize maintenance costs and maximize availability of railroad rolling stock, according to Miss Gertrude Patterson.

Miss Patterson, head of lubricating oil analysis for the Denver & Rio Grande Western Railroad at their research laboratory in Denver, spoke at the first regular meeting, Oct. 5th.

She is an expert in spectrographic analysis and has a highly developed knowledge of diesel and gasoline engine performance. Her work for the railroad is largely concerned with the large diesel-electric locomotives employed by D & R G.

The D & R G is pioneering in oil analysis to determine maintenance needs. "We have probed deeper into this subject than any of the other railroads out of necessity, since the severest of operating conditions are experienced in the Rocky Mountain region, combining long hauls with steep grades," Miss Patterson said.

Oil analysis was first instituted by the D & R G during the war when an extreme diesel mechanic shortage came about. Something had to be done to aid the understaffed force in determining where engine trouble existed without going through the time-consuming and expensive proceedure of tearing the engine down.

Today, through oil analysis, it is possible to detect even slight irregularities within the engine in time to prevent major breakdowns by minor and relatively inexpensive adjustments, Miss Patterson explained.

The three major tests run on oil as outlined by the speaker are the flash test, the filter test, and the spectrographic test.

The flash test is made on a half-pint sample of oil. By special apparatus the oil is heated, and the flash point observed. If the flash point falls a certain amount below the normal, dilution is present and the crankcase is drained.

"The flash test is 95% insurance against crankcase fire and explosion," Miss Patterson said, "and very worthwhile considering that a railroad in the East recently sustained damages of $60.000 due to an explosion of this sort."

The filter test is performed at regular intervals by the ASTM percipitation method, employing the use of a centrifuge to reveal excessive dirt or other foreign matter that would warrant investigation. If 0.5% dirt is present in the oil, the crankcase is drained.

The spectrographic test is made by reducing a sample of the oil to ash and arcing the ash in a spectrograph. By comparing this sample with lines of spectrum taken from a sample of a properly running engine the analysis is made.

"The spectrographic test gives con-

clusive evidence if certain abnormalities exist," Miss Patterson explained. "For example, if lead appears in the analysis, abnormal dilution is present. Iron, tin, or aluminum appearing in definite degrees will indicate excessive piston or ring wear. If copper shows up unfavorably, the main bearing isn't being lubricated properly, and the presence of silver will indicate trouble in the silver wristpin bearing."

The experience data used for comparison were compiled by the laboratory after years of research and study of test engines, Miss Patterson said. The test engines were run as far as 100,000 miles, using a lubricating oil of known constituency, which was sampled regularly, after which periods the engines were torn down and parts studied for wear, corrosion, and pitting.

—L. E. Johnson, Field Editor

New Members Qualified

These applicants qualified for admission to the Society between Sept. 10, 1949 and Oct. 10, 1949. Grades of membership are: (M) Member; (A) Associate; (J) Junior; (Aff.) Affiliate; (SM) Service Member; (FM) Foreign Member.

Atlanta Group

Linton S. Bethea (A), Zack T. Layfield, Jr. (A), Marion L. Weaver (A).

Baltimore Section

Ilia I. Islamoff (M), Carl Kenneth Kummerlowe (J).

British Columbia Section

Harry Robert Borrie (J), Frank Hepplewhite (A).

Buffalo Section

Robert W. Falconer (M), Burt James Finley (A).

Canadian Section

Fred Russell Hazelton (A), Albert (Bert) Edward Jagger (M), Ernest Albert Middleton (A), Reginald Lyle Miller (M), James Ray Montgomery (A), William John Scarlett (J), William H. Wright (A).

Central Illinois Section

Harvey M. Sibrel (M).

Chicago Section

Elmer Rudolph Bartosek (J), Richard T. Burnett (M), Noel R. Corder (J), Milo F. Denick (M), E. W. Hoogstra (A), James Joseph Kotlin (J), S. I. MacDuff (M), Paul Mongerson, Jr. (J), Arthur T. Pope (J), C. L. Sadler (M), Philip H. Sanders (M), Robert L. Shallenberg (M), Ludwig T. Stoyke (M).

Cincinnati Section

William Berliner (J), Merton B. Briggs (M), Jack P. Pedersen (A).

Cleveland Section

Howard M. Gammon (J), Harold A. Hudachek (J), David T. Muckley (A), William Elmer Newton (J), William J. Pankuch (A), C. F. Sudman (M), Forrest W. Sward (J).

Colorado Group

Donald L. Giddings (A).

Dayton Section

Thomas J. Borgstrom (J), William W. Leeper (A), William J. Short (SM), James Kenneth Thompson (J).

Detroit Section

Charles W. Adams (M), Harry P. Beally (J), Clare P. Briggs (A), Bruce

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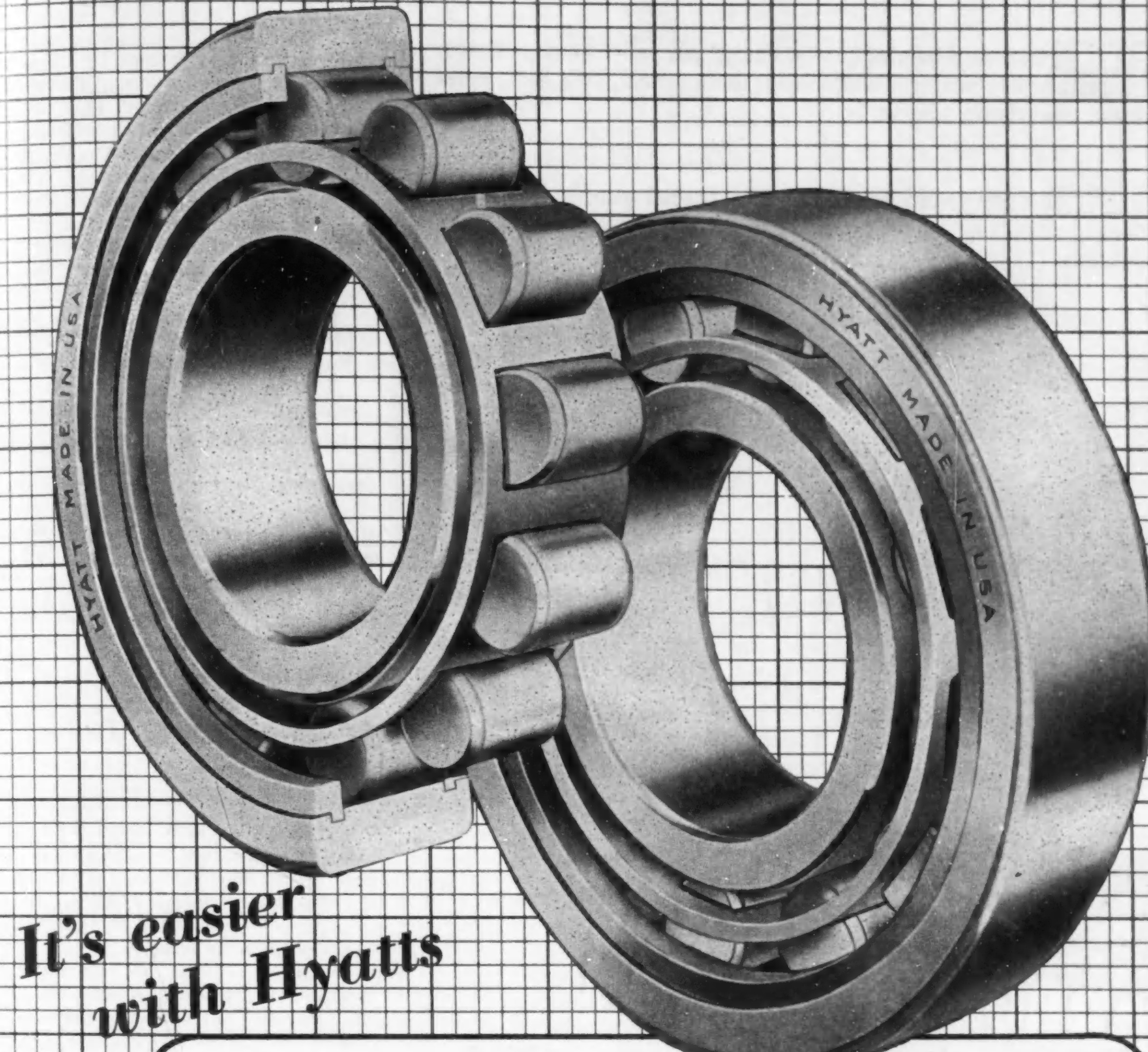
HYATT MADE IN USA
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Campbell (A), Harley F. Copp (J), Joseph F. Csernai (A), Charles M. Daniels, Jr. (A), Robert H. Duff (M), Donald T. Ellis (J), James Henderson (A), Stanley R. Hood (A), Vincent C. Judd (M), George Kalon (SM), Charles Stauffer Keller (J), Richard D. Kelly (A), Kenneth Peter Kirchoff (J), Witold Malecki (M), Robert M. Morkin (J), Charles Richter, Jr. (A), George Paul Vest (A).

Indiana Section

Eugene Phillip Welcher (J).

Kansas City Section

John Edward Ferguson (J).

Metropolitan Section

Morris Harris Alpert (J), Wilbur J. Bossome (J), Clarke F. Carey (J), John S. Conant (M), George J. Glaser (A), Charles A. Jackson (A), Dileep Nilkanth Kashalkar (J), George S. Tobias (M).

Milwaukee Section

Donald H. Dechant (J), Alfred William Schaper (M), Ferdinand E. Svanoe (A).

Mohawk-Hudson Group

W. Clement Palin, Jr. (J).

New England Section

William F. Hagen (J).

Northern California Section

Joseph Y. Chinn (J), Carl B. Engstrom (M), Donald T. Green (A), Martin Smolak (M), Harry James Van Auken (A), Frederic R. Watson (M), C. H. Wiget (M).

Northwest Section

Hilton H. Applegate (A), George M. Shrum (A).

Oregon Section

G. Douglas Hood (A).

Philadelphia Section

Robert Burke Benham (J), Ralph W. Benson (M), Charles Herbert Koch, Jr. (M), R. Philip Luce, Jr. (M), Laurence J. Test (M).

Pittsburgh Section

Frederick B. Kruger (J).

Southern California Section

Allen S. Baker (M), P. N. Fitzgerald (A), Paul L. Garver (M), Cecil Z. Green (A), Frank Mayer (M), Frank W. Murphy (M), Arthur H. Smith (A), Eugene F. Ward (A), George Harry Windsor (M).

Southern New England Section

John C. Mertz (M).

Syracuse Section

William A. Mader (M).

Texas Section

J. Swayne Cummings (A).

Twin City Section

William N. Blatt (J).

Washington Section

George C. Nield (J).

Outside of Section Territory

Robert Lewis Ednie (J), Kenneth Glade (SM), Fred V. Hamm (A), Ronald Frank Mower (J), Bernard A. Peskin (J), John Louis Wantland (J).

Foreign

Frederick E. S. Hudspeth (FM), England; John Stephen Langton (FM), England; C. H. Latimer-Needham (FM), England; Patrocle John Yangos (FM), Greece; Victor Gerald Young (A), England.

Applications Received

The applications for membership received between Sept. 10, 1949, and Oct. 10, 1949 are listed below.

Atlanta Group

Zack T. Layfield.

British Columbia Section

John H. Allan, Conrad A. Anderson.

Baltimore Section

Robert P. Gaston, Jr., Eric Albert Meyer, Charles F. Schwarz, James C. Scott.

Buffalo Section

John Dillon Walsh.

Canadian Section

George G. Ingles, Herbert Arthur Potton.

Chicago Section

C. V. Chappelle, Wesley H. Day, Stanley David Diamond, Robert P. Everett, Russell B. Hagen, Stephen Michael Hitchcock, Bruce H. Lundgren, John William Mansfield, Francis E. McDonald, Frederick Arthur Pitschke, George N. Schoonover, Mitchell H. Siskin, Howard W. Sonderman, Richard E. Wilkinson, Erwin E. Ziemann.

Cleveland Section

Daniel A. Canute, Robert George Chown, Carroll C. Colby, William Ralston Cox, Kenneth Wallace Cunningham, Jr., William Ellis Davidson, Louis H. Gegenheimer, Christian

George Goohs, Albert K. Hannum, Frank F. Hofstatter, Raymond J. Karabinus, John T. Korbuszewski, William G. Laffer, Oren Roy Linger, W. H. McKenzie, Fred Nevar, Abraham Schnapf, Millard G. Wettrich, John Norton Wholean, George Gene Younger, Donald V. Ziliox.

Dayton Section

Truman Gray Foster, Fred A. Irwin, Frederick A. Kondrotas, Franklin Cameron Lindsay, Robert L. Nathan.

Detroit Section

Julius Alberani, Robert Lynch Allen, Robert Anderson, Clarence G. Bauer, George William Betker, Jr., L. B. Billings, Ralph C. Bird, Jr., Edward C. Bockstahler, Jr., Stewart P. Bower, Robert W. Bristol, Basil E. Brown, Vincent A. Buck, Clarence R. Burdick, John William Clark, P. K. Coe, Sam Dabich, William J. David, Robert F. D'Haem, Charles E. Fiske, Todd W. Fredericks, John Ross Gretzinger, Byron A. Fay, Jr., George Edward Fosdick, Walter Hartung, Terry Hudyma, Derald Charles Katterman, Clayton W. Kerr, Herbert H. Kietzer, Nelson Wilton Kunz, James D. Leslie, Harold H. Miller, Donald H. Monson, David Palmer Moore, Arlinn H. Myers, George H. Primeau, C. B. Quillian, John B. Richardson, Louis Joseph Sabatini, Darrel R. Sand, Henry F. Schebor, Albert P. Schweizer, William Haven Smith, Robert H. Spahr, Jr., Frank A. Veraldi, James E. Vincenty, Robert A. Vogelei, Charles G. L. Walker, Major Rex Harry White, Jr., John H. Wilkinson, Jr., John G. Wright, E. E. Zimmerman.

Hawaii Section

Clarence C. Montgomery.

Indiana Section

Lowell A. Black, Jr., LeRoy V. Bradnick, Arthur W. Davis, John P. Krebser, Robert Frank Lay, E. A. Richards, Allen Randolph Stokke, Karl F. Wacker, Morris H. Wilson, Bradley A. Woodhull.

Metropolitan Section

Henry F. Capell, Gordon W. Duncan, Howard A. Grant, August B. Grillon, Hyman Hauser, Samuel H. Jackson, Bernard J. Kappler, Donato Masucci, Jerome Meyerson, James Edwin O'-Hara, Michael Vincent Piccniello, Chester J. Ramatowski, Bernard D. Rivin, Robert William Schubert, George Henry Schwarz, Thomas L. Shepherd, Howard O. Spaulding, Freddy Zaal.

Mid-Continent Section

William P. Barnes, Bill Eugene Council, Charles Joel Mauck, Lewis Adair Payne, Robert W. Young.

Milwaukee Section

Forrest D. Gunderson, Jr., Bernard Arthur Kloehn, Ross Clement Loving-

ton, George Lundin, Robert H. Wilke, Neal Carlyle Wogsland.

New England Section

Leslie L. Beeten, Bernard A. Fossa.

Northern California Section

Francis H. Bradford, Edward M. Harris, Joseph H. Kalakian, Robert Angus McDonald, B. Clinton Merithew, Robert F. Peterson, William Britt Roberts, Fred William Schmidt.

Northwest Section

David H. Farley, Edwin Lee Johnson, Robert G. Loser.

Oregon Section

Harry A. Dozier, David G. Kincaid, W. F. Romp, Sr.

Philadelphia Section

Malcolm Fox, Wiliam D. Gorman, Otis W. Marshall, Jr., Morris S. Ojalvo.

Pittsburgh Section

James F. Drake, Jr., Edward J. Kenna, William F. McClellan, Harry H. Takata.

St. Louis Section

Charles Franklin Dunlap, John M. Hartman, Ernest Edward Kornya.

Southern California Section

Harold W. Bailey, Allan G. Bell, Don B. Black, W. F. Cronin, James William Dawson, Mitchell O. Dion, Norbert Stark Gurell, Robert D. Koontz, Raymond F. Mahood, Donald S. Marden, Reynolds J. Mitchell, Roy J. Porter, Joaquin A. Saavedra, Edwin J. Schmidt, Jack Elliott Scott, Leo M. Shanahan, Gerald H. Smith, Robert E. Strasser, Mehmet Ayhan Turkkan, Joseph B. Violette, William Marmaduke Withrow.

Southern New England Section

John L. Cochran, Charles A. Peek, Jr., Alexander A. Zwierzchowski.

Texas Section

R. P. Akey, William B. Purnell, Leslie Alfred Ratliff, Jr., Joseph R. Shewitz, Constantine M. Xeros.

Virginia Section

Robert E. Hale, Jr.

Washington Section

David S. Graham, Simon Papazian, James E. Zulick.

Wichita Section

Wilbur C. Weissmann.

Williamsport Group

Merrell R. Fenske, Harry Guttman, Joseph Haag.

Outside of Section Territory

Donald R. Basham, John J. Daus, Jr., Joseph P. Duggan, George Rex Edwards, Horace S. Jeffrey, John L. Korleski, Jack L. LaVern, Henry W. Pfeifer, Vincent George Polovkas, Armin R. Reinert, Jr., Jack L. Roberts, William F. Smith, Jr., Edward M. Stiles.

Foreign

Attilio Bonesi, Italy; Hubert Dietrichs, Germany; James H. Grimes, Sau'di Arabia; Lawrence Derek Watts, England; David Fiarweather Young, South Africa.

HAVE YOU Changed Your Address?

So that your SAE mail will reach you with the least possible delay, please keep SAE Headquarters and the Secretary of your local Section or Group advised of any changes in your address. Such notices should be sent to:

1. Society of Automotive Engineers, Inc., 29 West 39th St., New York 18, N. Y.

2. The Secretary or Assistant Secretary of your Section or Group at the addresses listed below:

Baltimore
Ward L. Bennett, Baltimore Transfer Co., Monument & Forrest Sts., Baltimore 2, Md.

British Columbia
Burdette Trout, Suite 6, 1436 Pendrill St., Vancouver, B. C., Can.

Buffalo
Robert W. Morgan, 32 Windover, Hamburg, N. Y.

Canadian
Frank G. King, Canadian Automotive Trade, MacLean-Hunter Publishing Co., Ltd., 481 University Ave., Toronto 2, Ont., Can.

Central Illinois
James C. Porter, 120 Isabel Ave., Peoria, Ill.

Chicago
Floyd E. Ertsman, Chicago Section, SAE, 1420 Fisher Bldg., 343 S. Dearborn St., Chicago 4.

Cincinnati
B. E. Bennett, 55 Park Ave., Elsmere, P. O. Erlanger, Ky.

Cleveland
(Miss) C. M. Hill, 7016 Euclid Ave., Cleveland 3, Ohio

Dayton
Fred Schwenk, Fred's Auto Hospital, 1522 Xenia Ave., Dayton 10, Ohio

Detroit
(Mrs.) S. J. Duvall, Detroit Office, SAE, 100 Farnsworth Ave., Detroit 2, Mich.

Hawaii
Fred Hedemann, 2110 A Kakela Pl., Honolulu, T.H.

Indiana
Charles K. Taylor, L.G.S. Spring Clutch Corp., Indianapolis 6, Ind.

Kansas City
John P. Dranek, Box 304, Grandview, Mo.

Metropolitan
(Miss) J. A. McCormick, Society of Automotive Engineers, 29 West 39th St., New York 18, N. Y.

Mid-Continent
Delton R. Frey, 1427 S. 20th St., Chickasha, Okla.

Milwaukee
Charles H. Duquemin, Le Roi Co., 1706 S. 68th St., Milwaukee 14, Wis.

New England
Edward G. Moody, Ed. G. Moody, Inc., 680 Columbus Ave., Boston 20, Mass.

Northern California
Morris H. Pomeroy, P. O. Box 427, Burlingame, Calif.

Northwest
Alfred P. Nelsen, Wagner Electric Corp., 1918 First Ave., S. Seattle 4,

Oregon
Edward A. Haas, 6334 N. Missouri Ave., Portland 11, Oreg.

Philadelphia
Linn Edsall, 2301 Market St., Philadelphia 3, Pa.

Pittsburgh
Warren J. Iliff, Equitable Auto Co., 214 N. Lexington Ave., Pittsburgh 8, Pa.

St. Louis
August H. Blattner, Carter Carburetor Corp., 2840 N. Spring Ave., St. Louis 7, Mo.

San Diego
Jerome J. Wheeler, Jr., 7321 Oakwood Dr., Linda Vista, San Diego, Calif.

Southern California
Charles L. Fernau, Standard Oil Co. of Calif., 605 W. Olympic Blvd., Los Angeles 15, Calif.

Southern New England
Alexander M. Watson, 81 Foxcroft Dr., Manchester, Conn.

Spokane-Intermountain
Peter J. Favre, 10 W. Third Ave., Spokane 8, Wash.

Syracuse
Leslie R. Parkinson, 117 Harrington Rd., Syracuse, N. Y.

Texas
Ross A. Peterson, Texas Trade School, 1316 W. Commerce St., Dallas 8, Tex.

Twin City
Wayne E. Schober, 4620 Columbus Ave., Minneapolis, Minn.

Virginia
Robert P. Knighton, Esso Standard Oil Co., Broad & Hamilton Sts., Richmond 21, Va.

Washington
Hyman Feldman, 3222 M St., N.W., Washington, D. C.

Western Michigan
Thomas Reeves, Continental Motors Corp., Muskegon, Mich.

Wichita
Virgil W. Hackett, 143 N. Fountain, Wichita 8, Kans.